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Heating, Ventilating
and
Air-Conditioning Fundamentals

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William H. Severns

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Preface

The purpose of this book remains the same as that of the first edition—to present the essential fundamentals of heating, ventilating, and air conditioning. The material included is sufficient for undergraduate courses in the subject, as usually given for students in architecture, architectural engineering, and mechanical engineering.

Heating and air-conditioning methods are in a changing state of development. The fundamentals of the fields remain practically the same as those given in the first edition, but methods of application and the equipment available have changed during the past few years. The field of endeavor is now in a further state of development with many investigations and much research in progress. Great care has been taken to include in the material of this book the latest and most authentic data known at the time of its preparation which are readily applicable to the problems of the field.

Along with the applicable theory, examples of typical calculations have been included where they add clarity to the presentations. Representative commercial equipment has been illustrated and described; sufficient data pertaining to part of it are included so that a student may do some design work as part of his training.

Appreciation of the cooperation received from the manufacturers whose equipment is illustrated is hereby expressed. The permission granted by various individuals and organizations to use data developed and prepared by them has been of great assistance, and the right to use such material and its source have been indicated in all cases.

W. H. S.
J. R. F.

Urbana, Illinois
April 1949

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CHAPTER 1

ESSENTIAL DEFINITIONS, LAWS, AND PROPERTIES OF VARIOUS MEDIA

1. Heating. Losses of heat from an enclosed space may require the artificial maintenance of the air temperature within the enclosure at a value necessary either for the production of human comfort or for other reasons. The mere introduction of a sufficient quantity of heat into a building space may not produce either the comfortable, the necessary, or the desirable inside air temperatures when the outside weather conditions make such action imperative.

The problems of heating include: (1) the estimation of the load, (2) the procurement of heat from some source, (3) the selection of a medium to serve as a carrier of heat, (4) the conveyance of the heat-carrying medium to the proper locations by means of a feasible system, (5) satisfactory and successful distribution of the heat within the space to be heated, (6) regulation of the heat supply and air temperatures by various means, (7) the maintenance of the cycles of operation in the different phases of the system chosen, and (8) the reduction of unavoidable losses of heat to a feasible minimum.

2. Ventilation. In order to produce and to maintain either healthful, comfortable, or the necessary air conditions within a structure it may be necessary to introduce into and remove from the spaces of the building a definite amount of air per unit of time. This change of air, which can be accomplished by either natural or mechanical means, is termed ventilation. The air supplied may or may not be processed to improve its physical and chemical properties. The most successful and satisfactory systems of ventilation embody a part or all of the features of air conditioning.

3. Air Conditioning. The required treatment of, and the handling of air to produce, an atmospheric environment either conducive to human comfort or essential to the processing of materials in manufacturing operations is termed air conditioning. These two phases, comfort and industrial air conditioning, involve both summer and winter operation of plants for their production.

Atmospheric environment is affected more or less by a number of factors. Items included in appraising air are: its temperature, humidity (moisture content), motion, distribution, dust, bacteria, odors, toxic gases, and ionization. Air conditioning maintains simultaneous control of at least the first three items and preferably of all the factors mentioned. The function of an air-conditioning plant is to provide and to distribute air having satisfactory temperature, humidity, movement, and purity.

Winter treatment of air for comfort includes heating; humidification; cleaning, i.e., the removal of dust, dirt, smoke, fog, and bacteria by various means; sterilization; the absorption of odorous materials; and the dilution with clean fresh outside air either to reduce odor intensities or to minimize the dangers from toxic gases or both. The distribution of the air must be uniform with gentle motion in all parts of the occupied zone so that a sensation of comfort is produced for the greatest number of individuals. Summer air conditioning for comfort differs from winter conditioning in that the air is cooled and dehumidified.

Industrial air conditioning is often required during the storage of materials and products; it is often a necessity during some of the manufacturing operations of the following industries: brewing, chemicals and drugs, engines, electrical products, foods, furs, glass, precision instruments, machine shops, paints, petroleum products, paper, photographic materials, printing, rubber goods, steel, textiles, and tobacco in the form of cigarettes, cigars, and other materials. Proper air temperature, humidity, and cleanliness are of extreme importance in a number of the foregoing operations; otherwise the materials cannot be handled, or else the finished products are inferior or worthless. Industrial air conditioning frequently, but not always, produces comfort conditions for the workers employed.

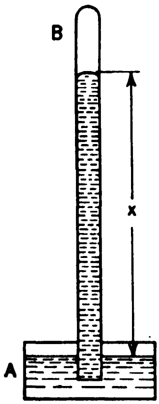

FIG. 1. Mercurial barometer.

4. Absolute Pressure. Fluids such as gases, liquids, steam, and vapors exert a force per unit of area upon the surfaces of their confining holders. Such a force, or the sum of such forces when two or more fluids are involved, is termed pressure. All measurements made by the use of ordinary pressure and vacuum gages are indications of the difference in pressure either above or below that of the atmosphere surrounding the gage. Atmospheric pressures are commonly measured by some form of a barometer, Fig. 1, which indicates the height, x, of a mercury column that exerts the same unit force at its base.

The standard barometric pressure, the average air pressure at sea level and 45° north latitude, is the unit pressure exerted by a column of mercury 29.921 in. in height which the sea-level atmosphere will support when the mercury is at a temperature of 32 F. Pure mercury at a temperature of 32 F weighs 0.4912 lb per cu in.; therefore the unit pressure represented by the standard barometric mercury column is $29.921 \times 0.4912 = 14.696$ psi (lb per sq in.).

The barometric pressure varies with the altitude and weather conditions. The algebraic sum of the gage reading and the barometric pressure, both being expressed in the same units, equals *the absolute pressure.*

5. Energy, Work, and Power. The ability to overcome resistance and thereby the capacity for doing work is termed *energy.* Common forms of energy are heat, mechanical, electrical, and chemical. Energy can be neither created nor destroyed, but its form can be changed under proper conditions. The common unit of heat energy is the *British thermal unit, Btu.* That of mechanical energy is the *foot-pound;* it represents the ability to lift a weight of one pound through a vertical distance of one foot. *Kinetic energy* is derived from the motion of a body. *Potential energy* accrues from the position of a body or parts of a body with respect to another or other bodies when either a force or forces tend to change the relative positions. *Internal energy,* as possessed by a substance, represents the sum of the kinetic and potential energies due to the motions and relative positions of its molecules with respect to each other.

Work results when a force acting through a distance overcomes a resistance. The unit of work is the foot-pound; it represents the action of a force of one pound through a distance of one foot. Work is independent of the time required for its accomplishment.

Power defines the rate at which work is done. Thus the mechanical horsepower represents the completion of 33,000 ft-lb of work in a period of one minute. The unit of electrical power is the *watt.* In non-inductive electrical circuits the product of the amperes of current flowing and the volts (electrical pressure) impressed represents watts. A mechanical horsepower is equivalent to 746 watts. A kilowatt is 1000 watts and is equal to 1.34 hp.

6. Heat. Irrespective of its physical condition any material possessing heat has its molecules in motion. This molecular motion increases with further heating and decreases as cooling occurs. Therefore heat is a form of molecular energy which may be produced by combustion of a fuel, friction, chemical action, and the resistance offered to the flow of electricity in a circuit.

Heat which produces a change of temperature (thermal head) in a body is known as *sensible heat;* that which is necessary to produce a change of state of a material at a constant temperature is known as *latent heat.* Latent heat has two forms: *heat of fusion* when a material passes from a solid to a liquid condition at a constant melting temperature and *heat of evaporation* when passage from a liquid to a vaporous state occurs under the existent conditions.

The amount of heat energy possessed by different substances at a given temperature varies with the material, its quantity, and its temperature. The basis of reckoning may be above any fixed reference point. The change in the amount of sensible heat of a body is dependent upon the weight of the substance involved, its thermal capacity (specific heat), and the temperature change. Heats of fusion and also heats of evaporation vary with the material and with the temperatures and pressures involved.

7. Temperature. Heat flows from a region of a higher temperature to one of a lower thermal head. Temperature alone does not indicate the quantity of heat involved, but it does give a measure of heat intensity and the ability of a body to transmit heat to other bodies.

Heating and air-conditioning temperature measurements may be made by glass thermometers, thermocouples, or pyrometers which may be calibrated in terms of either degrees Fahrenheit or degrees centigrade. Figure 2 shows a comparison of the scales of Fahrenheit and centigrade thermometers and the markings at the freezing and boiling points of pure water under atmospheric pressure at sea level. The fundamental range of the more commonly used Fahrenheit thermometer is 212 − 32 = 180 deg, and that of the centigrade thermometer is 100 − 0 = 100 deg. Therefore each degree centigrade is equal to $\frac{9}{5}$ deg F, and each degree Fahrenheit is equal to $\frac{5}{9}$ deg C. On the basis of the markings of 32 F and 0 C at the freezing point of pure water the conversions from one scale to another are

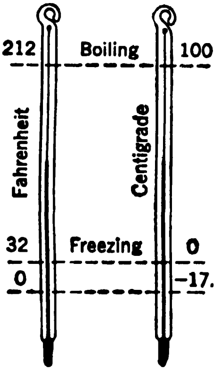

FIG. 2. Thermometer scales.

$$t_F = \frac{9}{5} t_C + 32 \qquad (1)$$

and

$$t_C = \frac{5}{9} (t_F - 32) \qquad (2)$$

8. Absolute Temperature. The total temperature as reckoned above a point known as the absolute zero of temperature is designated as the absolute temperature. Heat energy is lacking in a material at the absolute zero of temperature.

Experiments with the so-called perfect gases indicate that the volume changes of the gases, under constant pressure, when heated from 32 to 33 F and when cooled from 32 to 31 F amount to 1/491.6 of the original volume. If the change is uniform to the lowest temperatures the volume of a gas would become zero at 491.6 deg below 32 F. As all reckonings are made above and below the zero of the thermometer scale the absolute zero of the Fahrenheit scale is $32 - 491.6 = -459.6$ F. For most practical purposes a value of -460 may be used. When the absolute temperature is designated T and the temperature as indicated by a Fahrenheit thermometer is t the absolute temperature is expressed as

$$T = t + 459.6 \tag{3}$$

9. Heat Measurement. Any quantity of heat may be designated in terms of the British thermal unit, Btu, a mean value of which is taken as $\frac{1}{180}$ of the heat necessary to raise the temperature of one pound of pure water from 32 to 212 F.

10. Mechanical Equivalent of Heat. Mechanical work and heat energy are interchangeably convertible from one to the other in the ratio of 778.26 ft-lb to one Btu. For most calculations the mechanical equivalent may be used as 778 ft-lb. The heat equivalent of one mechanical horsepower-hour is $(33{,}000 \times 60) \div 778 = 2545$ Btu. Likewise the heat equivalent of a kilowatthour is found to be $(1000 \times 33{,}000 \times 60) \div (746 \times 778) = 3412$ Btu.

11. Energy Balances in Steady-Flow Processes. Heating and air-conditioning problems involve steady-flow processes which include the combustion of fuels; kinetic energy changes where the flow of fluids takes place; work done by engines, fans, and compressors; and heat transfer in exchange equipment. Inasmuch as energy cannot be destroyed but its form can be changed, engineers often account for the total energy entering and leaving a machine through which a working medium passes in a steady-flow process. Such a case is illustrated by Fig. 3, and in the accompanying discussion all subscripts designated by 1 and all those indicated by 2 refer to the conditions at points 1-1 and 2-2.

The quantity M represents the weight of the working medium flowing in pounds per second, v_1 and v_2 the velocities of flow in feet per second, Z_1 and Z_2 the distances in feet above some datum line, J

the mechanical equivalent of heat 778.26 ft-lb per Btu, and g the acceleration due to gravity taken as 32.174 ft per sec per sec. The kinetic energies $Mv_1^2/2gJ$ and $Mv_2^2/2gJ$ are due to the motion of the *medium as a whole. Under the headings of potential energy may be listed the chemical energies* C_1 *and* C_2*;* the potential energies MZ_1/J and MZ_2/J due to the position of or the condition of the medium; and the flow energies P_1V_1/J and P_2V_2/J, i.e., mechanical potential energy stored in volumes of V_1 and V_2 cu ft under pressures of P_1 and P_2 psfa (lb per sq ft abs). Entering into the analysis also are the internal energies U_1 and U_2 due to potential and kinetic energies of the molecules; the heat energy gain Q_i, and Q_o transferred by various means; and the mechanical work W_i done upon the operating fluid and the work W_o effected by it in passing through the apparatus. Net values of Q are considered as positive when heat is absorbed by a substance and negative if heat is rejected by it. Net work W is used as positive when done by the medium and negative when it is done upon the medium. When an analysis of a steady-flow process is made care must be taken to express everything involved either in terms of foot-pounds or in heat units, Btu, per unit of time, which preferably is one second.

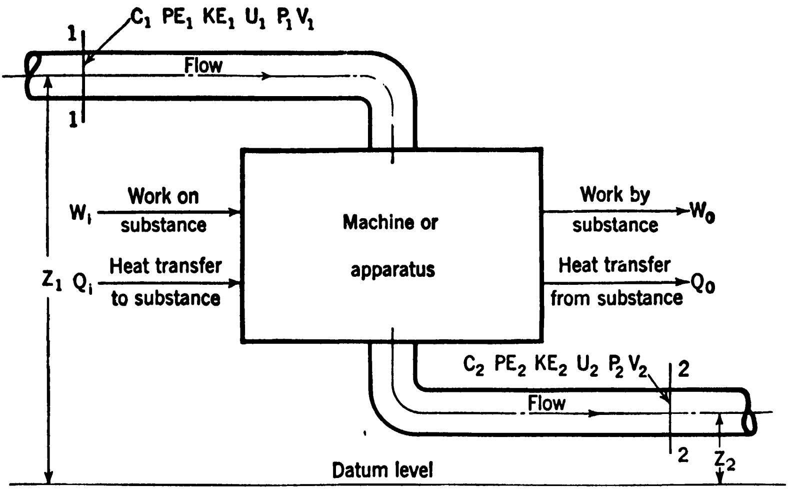

FIG. 3. Energy diagram for steady-flow analysis.

Expressed in terms of heat units per second the energy balance for a steady-flow condition is

$$C_1 + \frac{MZ_1}{J} + \frac{Mv_1^2}{2gJ} + U_1 + \frac{P_1V_1}{J} + Q$$
$$= C_2 + \frac{MZ_2}{J} + \frac{Mv_2^2}{2gJ} + U_2 + \frac{P_2V_2}{J} + \frac{W}{J} \quad (4)$$

In most cases parts C_1, MZ_1/J, C_2, and MZ_2/J may be ignored. The kinetic energies may be of importance in calculations. The quantities $U_1 + P_1V_1/J$ and $U_2 + P_2V_2/J$ each represent a term known as enthalpy which is discussed in the following article.

12. Enthalpy. The important forms of energy possessed by a fluid entering and leaving heating and air-conditioning apparatus are: kinetic energy due to the velocity of flow, internal energy, and flow (PV) energy. Inasmuch as the change of kinetic energy, $M(v_2^2 - v_1^2)/2gJ$ is generally small this quantity may usually be ignored, but at any point in heating and air-conditioning apparatus the internal energy and the flow energy are of great importance. The total heat of a material can be reckoned correctly only above a base temperature of 0 deg F abs, which is not always convenient. Enthalpy, which is the sum of the internal energy and the flow energy at any point, can be conveniently reckoned above 32 F for water and steam and 0 F for air. A change of enthalpy represents heat, Btu, added or abstracted in a process. The amount of heat added or abstracted is the common requirement in many heating and air-conditioning computations so that initial and final enthalpies may be used as conveniently as initial and final total heats might be. Enthalpy h per pound of any fluid is

$$h = u + Pv/J \quad (5)$$

where u = internal energy, Btu per lb.
P = total pressure, psfa.
v = specific volume of fluid, cu ft per lb.
J = mechanical equivalent of heat, 778 ft-lb per Btu.

13. Specific Heats. The thermal capacity or specific heat of a substance is the amount of heat added to or taken from a unit of weight of a material to produce a change of one degree in its temperature. In this work specific heats will be used as Btu per pound per degree Fahrenheit.

All specific heats vary with a change of temperature, and distinctions must be made between the true or instantaneous and the mean specific heat. The instantaneous specific heat is the amount of heat that must be added or abstracted at any definite temperature to produce a temperature change of one degree per unit weight of the material. The mean specific heat, over a given temperature range, is the average amount of heat required to produce a change of one degree of temperature per unit weight of the substance.

Specific heats of gases and vapors are also dependent upon the conditions of maintenance, i.e., either constant pressure or constant volume. The specific heat at constant pressure c_p is greater than the specific heat at constant volume c_v. When volume changes are produced at constant pressure, by temperature changes, external work is done, and the heat equivalent of the work is reflected in the specific heat c_p at constant pressure. The temperature changes of gases and vapors at constant volume produce pressure changes, but as no work is done the value of the specific heat at constant volume is always less than that for constant pressure by the amount of the heat equivalent of the external work done under constant-pressure conditions. There is little difference in the specific heats of liquids and solids at constant pressure and those at constant volume.

The specific heats of a few materials for various temperature ranges are given by the data of Table 1.

TABLE 1

MEAN SPECIFIC HEATS

Solids, Liquids, and Gases

Material	Temperature Range, Deg F	Specific Heat	Material	Temperature Range, Deg F	Specific Heat
Air					
Constant pressure	32–212	0.2401	Glass	32–212	0.163
Constant volume	32–212	0.173	Gypsum	32–212	0.259
Aluminum	61–579	0.225	Mercury	32–212	0.033
Asbestos	32–212	0.200	Oak	32–212	0.570
Brass	68–212	0.104	Petroleum	32–212	0.500
Brick	32–212	0.220	Pine	32–212	0.670
Cast Iron	68–212	0.119	Soil	32–212	0.440
Cinders	32–212	0.180	Steel	32–212	0.117
Concrete	32–212	0.156	Stone	68–208	0.208
Copper	59–540	0.095	Water	32–212	1.00
Cork	32–212	0.485	Zinc	68–212	0.093

The sensible heat added to or abstracted from a material is

$$H = cW(t_2 - t_1) \tag{6}$$

where H = heat, Btu.
c = the proper specific heat.
W = weight of the material, lb.
t_1 = lower temperature of the material, deg F.
t_2 = higher temperature of the material, deg F.

14. Dissemination of Heat. The passage of heat from regions of higher to those of lower temperature may be either by conduction, by convection, by radiation, or by combinations of the foregoing ways.

15. Conduction. The transfer of heat by conduction is accomplished by heat flow from one particle to another particle of a material and occurs in any direction where a lower temperature is being maintained. An example is furnished by hot and cold bodies in contact with each other. The rate of flow varies with the character of the material, the distance through which the flow takes place, and the temperature difference maintained. Some materials permit the rapid transmission of heat by conduction and soon become nearly uniform in temperature. Other materials, which serve as insulators, resist the flow of heat by conduction and may have great temperature differences existing within them.

The thermal conductivity k is the amount of heat, in Btu per hour, transmitted per square foot of area, per inch of thickness of the material, per one degree Fahrenheit of temperature difference. The total heat transmitted by conduction per hour is expressed as

$$H = \frac{k}{x}(t_2 - t_1)A \tag{7}$$

where H = heat, Btu per hr.
k = thermal conductivity.
x = thickness of material, in.
t_1 = temperature of the cooler surface or section, deg F.
t_2 = temperature of the hotter surface or section, deg F.
A = area involved, sq ft.

Inspection of equation 7 reveals that the thicker the material, of a given conductivity, the smaller will be the amount of heat conducted at a unit area with a fixed temperature difference.

16. Convection. A transfer of heat takes place as a fluid (liquid or gaseous) flows over the surface of a colder or a hotter body. The medium, set in motion by either natural or mechanical means, either loses or receives heat, depending upon its temperature at the point of contact relative to that of the surface over which it passes. Heat transfer by conduction takes place between the molecules of the moving agent and the material having the surfaces where the two are in contact. The transfer of heat by the carrying medium then becomes the process of convection.

An example of heat transfer by convection may be seen when water is heated in a pan placed on the top of a stove. Heat received from a burning fuel passes by conduction through the metal walls of the pan to the water molecules in contact with the water-heating surfaces. Distribution of the heat energy, by the process of convection, throughout the water takes place as natural circulation of the water is promoted by its changes of density.

Heat transfer by convection is independent of the nature of the surfaces over which the conveying medium moves. The amount of heat transferred per unit of time is affected by the velocity and nature of the moving medium, the area and form of the surfaces rubbed, and the difference between the temperature of the moving fluid and that of the surfaces rubbed.

17. Radiation. Heat dispersed by radiation passes through space in straight lines following the laws of the travel of light. Air and other gases, unless they contain moisture or dust particles, are not warmed directly by radiant heat. Some materials are absorbers, and others are reflectors of radiant heat. Materials generally have the power of emitting the same amount of radiant heat that they will absorb under reversed conditions of temperature difference. A few materials have the property of allowing radiant heat to pass through them without absorbing any of the heat transmitted. Other materials diffuse radiant heat in all directions without regularity or uniformity.

Transmission of radiant heat varies inversely as the square of the distance between the source of radiation and the absorbing surface. Unless there are re-entrant angles of the surface to produce interference in the heat rays, the amount of heat radiated in a given time, per unit of area, is independent of the form and extent of a heated body. Heat emission by radiation is dependent upon the nature of the radiating surface, its absolute value of temperature, and the temperature differences maintained. The heat transmission varies as the difference of the fourth powers of the absolute temperatures of the radiating and the receiving bodies.

The general expression for the transmission of radiant heat per unit of surface per hour according to the Stefan-Boltzmann law is

$$H = K(T_1^4 - T_2^4) \tag{8}$$

where H = heat radiated per unit of surface, Btu per hr.

K = a coefficient which includes a radiation constant, a relative-position factor for the emitting and the receiving surfaces, and absorption and emission characteristics for the two.

T_1 = temperature of the heat radiating surface, deg F abs.

T_2 = temperature of the heat receiving surface, deg F abs.

18. Expansion and Contraction. The linear dimensions and volumes of nearly all materials are effected by heat changes and the consequent changes of temperature. In engineering designs, such as the layout and installation of steam and hot-water pipe lines, cognizance must be taken of the movement produced by expansion and contraction. If provisions are not made for such movement the piping may be thrust out of alignment or the fittings ruptured by the excessive stresses produced. Such contingencies are avoided by the use of expansion joints, pipe bends, and loops.

The amount of linear expansion or contraction produced is dependent upon the linear length of material involved, its coefficient of expansion, and its temperature change in degrees. The coefficient of linear expansion is the linear change per inch of length of a material per one degree of temperature change. The coefficient of linear expansion varies with the temperature; therefore mean values are usually stated for given temperature ranges.

Table 2 gives the coefficients of linear expansion for several metals used in heating and air-conditioning plants.

TABLE 2

COEFFICIENTS OF LINEAR EXPANSION

Metal	Coefficient, In. per Deg F per In.	Metal	Coefficient, In. per Deg F per In.
Aluminum	0.000,012,33	Steel, soft-rolled	0.000,006,30
Brass, rolled	0.000,010,72	Wrought iron	0.000,006,73
Cast iron	0.000,005,89	Zinc	0.000,016,53
Copper, drawn	0.000,009,26		

The linear expansion or contraction is

$$E = C_e L(t_2 - t_1) \tag{9}$$

where E = change of linear length, in.

C_e = coefficient of linear expansion.

L = original length, in.

t_1 = lower temperature of the material, deg F.

t_2 = higher temperature of the material, deg F.

Volumetric expansion is of importance in fluids, such as water in hot-water heating systems, where compensation must be provided either in the form of expansion tanks or relief valves. For water between 32 and 212 F the coefficient of cubical expansion is 0.000,243 per deg F.

19. Gas Laws. Perfect gases are those that can be liquefied only with great difficulty and which closely follow the laws of Boyle and

Charles. Air, oxygen, hydrogen, nitrogen, and helium are often considered as nearly perfect gases.

Boyle's law states that, when the temperature is held constant, the volume of a given weight of a perfect gas varies inversely as the absolute pressure. *Charles' law* specifies that, when a perfect gas receives heat at constant volume, the absolute pressure varies directly as the absolute temperature. The following single equation satisfies both of the foregoing statements.

$$PV = WRT \tag{10}$$

where P = absolute pressure, lb per sq ft.
V = volume of gas, cu ft.
W = weight of gas, lb.
R = gas constant, ft-lb per lb per deg.
T = absolute temperature, deg F.

Equation 10 may also be written as $WR = PV/T$ and for a constant weight W of a given gas

$$\frac{PV}{T} = \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \tag{11}$$

Equations 10 and 11 are useful in heating and air-conditioning calculations where pressure, volume, and temperature changes occur in air and other gases.

The volume of a mole weight (a weight, in pounds, equal to the molecular weight m) of any gas is 358.65 cu ft when its pressure is 14.7 psia and its temperature 32 F. When W is equal to m, based on equation 10, the following statements are true: $R = PV/WT = PV/mT = B/m$ in which B, a constant, is PV/T or $(14.7 \times 144 \times 358.65) \div (459.6 + 32) = 1544$ ft-lb per deg F. The molecular weight of oxygen, O_2, is $16 \times 2 = 32$, and the value of R for oxygen is $1544 \div 32 = 48.25$ ft-lb per lb per deg F.

Values of R for a few gases are given in Table 3.

TABLE 3

GAS CONSTANTS

Gas	R	Gas	R
Air	53.35	Nitrogen	55.01
Carbon dioxide	35.10	Oxygen	48.25
Hydrogen	765.36	Water vapor	85.71
Methane	96.31		

For air at atmospheric pressure equation 10 may be changed to the following form to give the weight of one cubic foot, or the density

d_a, of dry air at any temperature and pressure. W equals d_a when V is equal to one cubic foot.

$$d_a = \frac{PV}{RT}$$

$$= \frac{144 \times 0.4912 \times \text{barometric pressure, inches of mercury} \times 1}{53.35(459.6 + t)}$$

$$\text{Hence, } d_a = 1.3258 \times \frac{\text{barometric pressure, inches of mercury}}{(459.6 + t)} \qquad (12)$$

where t = air temperature, deg F.

Example. Find the density of dry air at a temperature of 60 F when the barometric pressure is 28.5 in. of mercury.

$$d_a = 1.3258 \times \frac{28.5 \times 1}{(459.6 + 60)} = 0.07272 \text{ lb per cu ft}$$

The specific volume v or cubic feet per one pound of gas is the reciprocal of the density, $1/d_a$, and is equal, in the foregoing case, to 1 ÷ 0.07272 = 13.75 cu ft per lb. The weights of dry air per cubic foot at various temperatures and a barometric pressure of 29.921 in. of mercury, as given in Table 8, were obtained by the use of equation 12.

20. Water. Pure water (H_2O) is the result of the chemical combination of hydrogen and oxygen in the ratio of 2 parts of the former to 16 parts of the latter by weight. The maximum density of pure water occurs at a temperature of 39.2 F; the freezing and boiling temperatures under an atmospheric pressure of 14.7 psia, alone, are 32 and 212 F respectively. The boiling temperature is made higher by increases of the absolute pressure. The volumes in cubic feet per pound and the weights per cubic foot for water at various temperatures as derived from Table 6, Properties of Saturated Steam, by J. H. Keenan and F. G. Keyes, are given by Table 4. The weights per cubic foot, at the various temperatures, are slightly different from those when the water pressure is held constant at 14.7 psia. The values given are satisfactory for most calculations.

21. Steam Generation. A sufficiently rapid transfer of heat to water held in an open or closed vessel produces changes in the physical state of the water. The effects are: increased vibration of the molecules, a rise in the temperature, and a decrease in density of the liquid. All these changes continue until the temperature at which boiling occurs is reached. The temperature at which boiling or vaporization occurs is controlled by the absolute pressure exerted upon the water.

The boiling temperature of pure water is definitely fixed for each pressure and is less at lower than at higher pressures.

TABLE 4

Properties of Water*

Temperature, Deg F	Boiling or Saturation Pressure, Psia	Volume, Cu Ft per Lb	Weight, Lb per Cu Ft	Enthalpy or Heat above 32 F, Btu per Lb	Temperature, Deg F	Boiling or Saturation Pressure, Psia	Volume, Cu Ft per Lb	Weight, Lb per Cu Ft	Enthalpy or Heat above 32 F, Btu per Lb
32	0.08854	0.01602	62.42	0.0	135	2.5370	0.01627	61.46	102.90
35	0.09995	0.01602	62.42	3.02	140	2.8886	0.01629	61.39	107.89
40	0.12170	0.01602	62.42	8.05	145	3.281	0.01632	61.27	112.89
45	0.14752	0.01602	62.42	13.06	150	3.718	0.01634	61.20	117.89
50	0.17811	0.01603	62.38	18.07	155	4.203	0.01637	61.09	122.89
55	0.2141	0.01603	62.38	23.07	160	4.741	0.01639	61.01	127.89
60	0.2563	0.01604	62.34	28.06	165	5.335	0.01642	60.90	132.89
65	0.3056	0.01605	62.30	33.05	170	5.992	0.01645	60.79	137.90
70	0.3631	0.01606	62.27	38.04	175	6.715	0.01648	60.68	142.91
75	0.4298	0.01607	62.23	43.03	180	7.510	0.01651	60.57	147.92
80	0.5069	0.01608	62.19	48.02	185	8.383	0.01654	60.46	152.93
85	0.5959	0.01609	62.15	53.00	190	9.339	0.01657	60.35	157.95
90	0.6982	0.01610	62.11	57.99	195	10.385	0.01660	60.24	162.97
95	0.8153	0.01612	62.03	62.98	200	11.526	0.01663	60.13	167.99
100	0.9492	0.01613	61.99	67.97	205	12.771	0.01666	60.02	173.02
105	1.1016	0.01615	61.92	72.95	210	14.123	0.01670	59.88	178.05
110	1.2748	0.01617	61.84	77.94	215	15.595	0.01673	59.77	183.09
115	1.4709	0.01618	61.80	82.93	220	17.186	0.01677	59.63	188.13
120	1.6924	0.01620	61.73	87.92	225	18.916	0.01681	59.48	193.17
125	1.9420	0.01622	61.65	92.91	230	20.780	0.01684	59.38	198.23
130	2.2225	0.01625	61.54	97.90	240	24.969	0.01692	59.10	208.34

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* From *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, John Wiley and Sons, Inc., New York, 1936.

22. Saturated Steam. Steam which exists at the temperature corresponding to its absolute pressure is defined as saturated. Saturated steam may be completely free from unvaporized water particles, or it may carry water globules in suspension. Thus, saturated steam may be in either a dry or a wet condition.

In the production of saturated steam the heat or enthalpy added to each pound of water is utilized in two stages. The first stage raises the temperature of the water from that of its initial condition to that which corresponds to the boiling point at the existing absolute pressure. The heat necessary to accomplish this is the *heat or enthalpy of the liquid.* This stage is reached with a comparatively small increase of volume. The second stage in the process of steam formation is the addition of an amount of heat necessary to convert the liquid at its boiling point into a vapor. The latter quantity is the *heat or enthalpy of evaporation or latent heat of vaporization.* The vaporization takes place at a constant temperature corresponding to the boiling point.

Properties of Dry and Saturated Steam as compiled by Professors J. H. Keenan and F. G. Keyes are given by permission in abridged form in Tables 5 and 6. The steam table data are based on a weight of one pound of water or steam, and the heats are reckoned above a base temperature of 32 F.

23. Enthalpy of Liquid. The amount of heat in Btu necessary to raise one pound of water from a temperature of 32 F to the temperature of boiling, at any given pressure, is the heat or enthalpy of the liquid. The enthalpy of the liquid per pound of water may be computed as

$$h_f = c_{pw}(t - 32) \tag{13}$$

where h_f = enthalpy of the liquid, Btu per lb.
c_{pw} = mean specific heat of the liquid.
t = vaporization temperature, deg F.

The amounts of heat energy added to the water, as enthalpy of the liquid, increase as the pressure and temperature rise. The rate of the increase in the amount of heat added is not uniform for equal increments of pressure rise but is variable, for the mean specific heat of water is not constant but varies with different temperature ranges. For all temperatures the most accurate results are obtained by use of the enthalpies of liquid given in Tables 5 and 6.

24. Enthalpy of Evaporation. The heat in Btu necessary to convert one pound of liquid into dry steam at the same temperature and pressure is the enthalpy of evaporation or the latent heat of vaporization h_{fg}. The numerical quantities for the latent heat of vaporization are given in Tables 5 and 6 in the "Enthalpy of Evaporation" column. As the absolute pressure of steam increases, the enthalpy of evaporation decreases, becoming zero at the critical pressure of 3206 psia.

The enthalpy of evaporation is made up of two parts, the external and the internal latent heats. The external heat of evaporation is the heat equivalent of the external work necessary to change the volume of one pound of water to the volume occupied by one pound of the dry steam formed at the existing pressure. The internal heat of evaporation is the heat equivalent, in Btu, of the internal energy necessary to overcome the molecular attraction of the water molecules at the vaporization temperature.

25. Total Enthalpy of Dry Saturated Steam. The total enthalpy of one pound of dry saturated steam is the sum of the enthalpy of the liquid and the enthalpy of evaporation:

$$h_g = h_f + h_{fg} \tag{14}$$

where h_g = total enthalpy above 32 deg F, Btu per lb.
h_f = enthalpy of the liquid, Btu per lb.
h_{fg} = enthalpy of evaporation, Btu per lb.

The enthalpy of dry saturated steam is dependent upon its pressure as shown by the numerical quantities given in Tables 5 and 6 under "Enthalpy of Saturated Vapor, h_g."

When use is made of equation 14 the values of h_f and h_{fg} are taken from either Table 5 or 6 for the absolute steam pressure corresponding to that of the steam under consideration. Tables 5 and 6 are not complete in that the "Properties of Dry Saturated Steam" are not given for all pressures. However, when the properties are not given in the tables for various desired pressures, numerical data may be obtained by interpolating between the data given for the pressures nearest to the pressure to be used.

26. Specific Volume and Density of Dry Saturated Steam. The specific volumes v_g, in cubic feet, occupied by one pound of dry saturated steam are given in the fourth column of Tables 5 and 6. The volume of one pound of steam varies inversely as the pressure.

The density d of dry saturated steam is the weight of steam in pounds per cubic foot. The density is the reciprocal of the specific volume v_g per pound and is

$$d = \frac{1}{v_g} \tag{15}$$

The density increases with pressure increases; thus the higher the pressure the greater the weight of the steam per cubic foot. The volumes v_f per pound of the liquid or the water are given in the third column of the steam tables, and the density may be computed by means of equation 15.

27. Quality of Steam. Saturated steam may or may not carry entrained water. Moisture present in steam may be due to several causes. Steam conveyed by piping may suffer a reduction in heat con-

TABLE 5

Properties of Dry Saturated Steam: Pressure Table*

Pressure, Psia	Temperature F	Specific Volume		Enthalpy			Pressure, Psia
		Sat. Liquid	Sat. Vapor	Sat. Liquid	Evaporation	Sat. Vapor	
p	t	v_f	v_g	h_f	h_{fg}	h_g	p
0.20	53.14	0.01603	1526.0	21.21	1063.8	1085.0	0.20
0.40	72.86	0.01606	791.9	40.89	1052.7	1093.6	0.40
0.60	85.21	0.01608	540.0	53.21	1045.7	1098.9	0.60
0.80	94.38	0.01612	411.7	62.36	1040.4	1102.8	0.80
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	1.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	2.0
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	3.0
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	4.0
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	5.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	6.0
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	7.0
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	8.0
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	9.0
10.0	191.23	0.01659	38.42	161.17	982.1	1143.3	10.0
11.0	197.75	0.01662	35.14	165.73	979.3	1145.0	11.0
12.0	201.96	0.01665	32.40	169.96	976.6	1146.6	12.0
13.0	205.88	0.01667	30.06	173.91	974.2	1148.1	13.0
14.0	209.56	0.01670	28.04	177.61	971.9	1149.5	14.0
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	14.696
16.0	216.32	0.01674	24.75	184.42	967.6	1152.0	16.0
18.0	222.41	0.01679	22.17	190.56	963.6	1154.2	18.0
20.0	227.96	0.01683	20.089	196.16	960.1	1156.3	20.0
22.0	233.07	0.01687	18.375	201.33	956.8	1158.1	22.0
24.0	237.82	0.01691	16.938	206.14	953.7	1159.8	24.0
26.0	242.25	0.01694	15.715	210.62	950.7	1161.3	26.0
28.0	246.41	0.01698	14.663	214.83	947.9	1162.7	28.0
30.0	250.33	0.01701	13.746	218.82	945.3	1164.1	30.0
35.0	259.28	0.01708	11.898	227.91	939.2	1167.1	35.0
40.0	267.25	0.01715	10.498	236.03	933.7	1169.7	40.0
45.0	274.44	0.01721	9.401	243.36	928.6	1172.0	45.0
50.0	281.01	0.01727	8.515	250.09	924.0	1174.1	50.0

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* Abridged from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, John Wiley and Sons, Inc., New York, 1936.

TABLE 5 (*Continued*)

Pressure, Psia	Temperature F	Specific Volume		Enthalpy			Pressure, Psia
		Sat. Liquid	Sat. Vapor	Sat. Liquid	Evaporation	Sat. Vapor	
p	t	v_f	v_g	h_f	h_{fg}	h_g	p
60.0	292.71	0.01738	7.175	262.09	915.5	1177.6	60.0
70.0	302.92	0.01748	6.206	272.61	907.9	1180.6	70.0
80.0	312.03	0.01757	5.472	282.02	901.9	1183.1	80.0
90.0	320.27	0.01766	4.896	290.56	894.7	1185.3	90.0
100.0	327.81	0.01774	4.432	298.40	888.8	1187.2	100.0
110.0	334.77	0.01782	4.049	305.66	883.2	1188.9	110.0
120.0	341.25	0.01789	3.728	312.44	877.9	1190.4	120.0
130.0	347.32	0.01796	3.455	318.81	872.9	1191.7	130.0
140.0	353.02	0.01802	3.220	324.82	868.2	1193.0	140.0
150.0	358.42	0.01809	3.015	330.51	863.6	1194.1	150.0
160.0	363.53	0.01815	2.834	335.93	859.2	1195.1	160.0
170.0	368.41	0.01822	2.675	341.09	854.9	1196.0	170.0
180.0	373.06	0.01827	2.532	346.03	850.8	1196.9	180.0
190.0	377.51	0.01833	2.404	350.79	846.8	1197.6	190.0
200.0	381.79	0.01839	2.288	355.36	843.0	1198.4	200.0
210.0	385.9	0.01844	2.183	359.77	839.2	1199.0	210.0
220.0	389.86	0.01850	2.087	364.02	835.6	1199.6	220.0
230.0	393.68	0.01854	1.9992	368.13	832.0	1200.1	230.0
240.0	397.37	0.01860	1.9183	372.12	828.5	1200.6	240.0
250.0	400.95	0.01865	1.8438	376.0	825.1	1200.1	250.0
260.0	404.42	0.01870	1.7748	379.76	821.8	1201.5	260.0
270.0	407.78	0.01875	1.7107	383.42	818.5	1201.9	270.0
280.0	411.05	0.01880	1.6511	386.98	815.3	1202.3	280.0
290.0	414.23	0.01885	1.5954	390.46	812.1	1202.6	290.0
300.0	417.33	0.01890	1.5433	393.84	809.0	1202.8	300.0

tent through unavoidable or other losses so that part of the latent heat or enthalpy of evaporation of the steam is given up and water particles are formed in the steam. The generation of steam may have been imperfect on account of poor design of the boiler, high rate of evaporation maintained, or poor boiler feedwater.

The dry steam content per pound of wet steam is the *quality* of the steam. The quality x of steam may be expressed either as a percentage, or as the weight of dry steam existing within a pound of wet steam.

TABLE 6

PROPERTIES OF DRY SATURATED STEAM: TEMPERATURE TABLE*

Temperature F	Pressure, Psia	Specific Volume		Enthalpy			Temperature F
		Sat. Liquid	Sat. Vapor	Sat. Liquid	Evaporation	Sat. Vapor	
t	p	v_f	v_g	h_f	h_{fg}	h_f	t
32	0.08854	0.01602	3306.0	0.00	1075.8	1075.9	32
34	0.09603	0.01602	3061.0	2.02	1074.7	1076.7	34
36	0.10401	0.01602	2837.0	4.03	1073.6	1077.6	36
38	0.11256	0.01602	2632.0	6.04	1072.4	1078.4	38
40	0.12170	0.01602	2444.0	8.05	1071.3	1079.3	40
42	0.13150	0.01602	2271.0	10.05	1070.1	1080.2	42
44	0.14199	0.01602	2112.0	12.06	1068.9	1081.0	44
46	0.15323	0.01602	1964.3	14.06	1067.8	1081.9	46
48	0.16525	0.01603	1828.6	16.07	1066.7	1082.8	48
50	0.17811	0.01603	1703.2	18.07	1065.6	1083.7	50
52	0.19182	0.01603	1587.6	20.07	1064.4	1084.5	52
54	0.20642	0.01603	1481.0	22.07	1063.3	1085.4	54
56	0.2220	0.01603	1382.4	24.06	1062.2	1086.3	56
58	0.2386	0.01604	1291.1	26.06	1061.0	1087.1	58
60	0.2563	0.01604	1206.7	28.06	1059.9	1088.0	60
62	0.2751	0.01604	1128.4	30.05	1058.8	1088.9	62
64	0.2951	0.01605	1055.7	32.05	1057.6	1089.7	64
66	0.3164	0.01605	988.4	34.05	1056.5	1090.6	66
68	0.3390	0.01605	925.9	36.04	1055.5	1091.5	68
70	0.3631	0.01606	867.9	38.04	1054.3	1092.3	70
72	0.3886	0.01606	813.9	40.04	1053.2	1093.2	72
74	0.4156	0.01606	763.8	42.03	1052.8	1094.1	74
76	0.4443	0.01607	717.1	44.03	1050.9	1094.9	76
78	0.4747	0.01607	673.6	46.02	1049.8	1095.8	78
80	0.5069	0.01608	633.1	48.02	1048.6	1096.6	80
82	0.5410	0.01608	595.3	50.01	1047.5	1097.5	82
84	0.5771	0.01609	560.2	52.01	1046.4	1098.4	84
86	0.6152	0.01609	527.3	54.00	1045.2	1099.2	86
88	0.6556	0.01610	496.7	56.00	1044.1	1100.1	88
90	0.6982	0.01610	468.0	57.99	1042.9	1100.9	90
92	0.7432	0.01611	441.3	59.99	1041.8	1101.8	92
94	0.7906	0.01612	416.2	61.98	1040.7	1102.6	94
96	0.8407	0.01612	392.8	63.98	1039.5	1103.5	96
98	0.8935	0.01613	370.9	65.97	1038.4	1104.4	98

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* Abridged from *Thermodynamic Properties of Steam*, by Joseph H. Keenan and Frederick G. Keyes, John Wiley and Sons, Inc., New York, 1936.

TABLE 6 (*Continued*)

Temperature F t	Pressure, Psia p	Specific Volume		Enthalpy			Temperature F t
		Sat. Liquid v_f	Sat. Vapor v_g	Sat. Liquid h_f	Evaporation h_{fg}	Sat. Vapor h_g	
100	0.9492	0.01613	350.4	67.97	1037.2	1105.2	100
105	1.1016	0.01615	304.5	72.95	1034.3	1107.3	105
110	1.2748	0.01617	265.4	77.94	1031.6	1109.5	110
115	1.4709	0.01618	231.9	82.93	1028.7	1111.6	115
120	1.6924	0.01620	203.27	87.92	1025.8	1113.7	120
125	1.9420	0.01622	178.61	92.91	1022.9	1115.8	125
130	2.2225	0.01625	157.34	97.90	1020.0	1117.9	130
135	2.5370	0.01627	138.95	102.90	1017.0	1119.9	135
140	2.8886	0.01629	123.01	107.89	1014.1	1122.0	140
145	3.281	0.01632	109.15	112.89	1011.2	1124.1	145
150	3.718	0.01634	97.07	117.89	1008.2	1126.1	150
155	4.203	0.01637	86.52	122.89	1005.2	1128.1	155
160	4.741	0.01639	77.29	127.89	1002.3	1130.2	160
165	5.335	0.01642	69.19	132.89	999.3	1132.2	165
170	5.992	0.01645	62.06	137.90	996.3	1134.2	170
175	6.715	0.01648	55.78	142.91	993.3	1136.2	175
180	7.510	0.01651	50.23	147.92	990.2	1138.1	180
185	8.383	0.01654	45.31	152.93	987.2	1140.1	185
190	9.339	0.01657	40.96	157.95	984.1	1142.0	190
200	11.526	0.01663	33.64	167.99	977.9	1145.9	200
210	14.123	0.01670	27.82	178.05	971.6	1149.7	210
220	17.186	0.01677	23.15	188.13	965.2	1153.4	220
230	20.780	0.01684	19.38	198.23	958.8	1157.0	230
240	24.969	0.01692	16.32	208.34	952.2	1160.5	240

28. Enthalpy of Wet Saturated Steam. The total enthalpy of a pound of wet saturated steam is less than that of a pound of dry saturated steam under the same pressure conditions. When the water is not completely vaporized, the heat carried as the heat of evaporation is less. The quality affects only the enthalpy of evaporation and does not change the enthalpy of the liquid at any given pressure. The quality factor must be applied only to the enthalpy of evaporation, and not to the enthalpy of the liquid or to the enthalpy of a pound of dry saturated steam. The enthalpy of a pound of wet steam is

$$h_g = h_f + xh_{fg} \tag{16}$$

where h_g = enthalpy of wet steam, Btu per lb.
x = quality of the steam, expressed as a decimal.
h_f = enthalpy of the liquid, Btu per lb.
h_{fg} = enthalpy of evaporation, Btu per lb.

29. Superheated Steam. The temperature and enthalpy of saturated steam, at any pressure, may be increased by the application of additional heat. When the steam temperature is above saturation temperature, corresponding to the steam pressure, the steam is superheated. The superheating process usually takes place in a separate piece of equipment.

TABLE 7

MEAN SPECIFIC HEATS OF SUPERHEATED STEAM*

Steam Pressure, Psia	Sat. Temp, Deg F	Actual Steam Temperature, Deg F						
		200	250	300	350	400	450	500
10	193.21	0.485	0.477	0.474	0.471	0.470	0.470	0.470
12	201.96		0.483	0.477	0.475	0.473	0.473	0.472
14	209.56		0.487	0.481	0.477	0.475	0.475	0.474
15	213.03		0.489	0.483	0.479	0.477	0.476	0.475
16	216.32		0.492	0.484	0.480	0.478	0.476	0.475
18	222.41		0.496	0.488	0.484	0.480	0.478	0.477
20	227.96		0.500	0.490	0.486	0.482	0.480	0.479
50	281.01			0.537	0.521	0.512	0.506	0.501
100	327.81				0.581	0.573	0.544	0.533
150	358.42					0.608	0.582	0.565
200	381.79					0.653	0.620	0.596
250	400.95						0.661	0.629
300	417.33						0.704	0.665
350	431.72						0.754	0.697
400	444.59						0.780	0.732
450	456.28							0.773
500	467.01							0.815

* Based on data from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, John Wiley and Sons, Inc., New York, 1936.

The enthalpy of a pound of superheated steam is made up of three items: (1) the enthalpy of the liquid; (2) the enthalpy of evaporation: and (3) the heat added as superheat. In equation form the foregoing is expressed as

$$h = h_f + h_{fg} + c_{ps}(t_{sh} - t) \tag{17}$$

where h = enthalpy of superheated steam, Btu per lb.
h_f = enthalpy of liquid, Btu per lb.
h_{fg} = enthalpy of evaporation, Btu per lb.
c_{ps} = mean specific heat of superheated steam at the temperature and pressure of superheat.
t_{sh} = temperature of the superheated steam, deg F.
t = temperature of saturated steam at the existing pressure, deg F.

The specific heat of superheated steam varies with the existent pressure and the amount of the superheating. The temperature difference $(t_{sh} - t)$ is the number of degrees of superheat and is indicative of the rise of the temperature above that of saturation conditions. Unless the mean specific heat of the superheated steam is known, the heat added as superheat cannot be computed from the temperature rise of the steam. Table 7 gives values of the mean specific heats of superheated steam, for various absolute pressures and degrees Fahrenheit steam temperature.

PROBLEMS

1. The steam pressure shown by a gage was 152 psi when the barometric pressure was 29.3 in. of mercury. What was the absolute pressure in pounds per square inch?

2. The absolute pressure of the steam in a heating boiler was 20 psi when the barometric pressure was 27.45 in. of mercury. Find the gage pressure of the steam in pounds per square inch.

3. A vacuum gage indicated that the pressure within a closed chamber was 26 in. of mercury less than the barometric pressure of 28.5 in. of mercury outside. Find the absolute pressure within the chamber in pounds per square inch.

4. Twenty-thousand watts of electrical power were supplied continuously each hour to operate the motors driving fans. Find the equivalents in terms of kilowatthours and horsepower-hours.

5. The output of a generator is 50 kw when operated at 50 per cent of its rating. How many foot-pounds of work per minute are equivalent to an output representing 100 per cent of its rating?

6. Express 50 F as degrees centigrade and 80 C as degrees Fahrenheit. What is the absolute temperature in each case expressed in degrees Fahrenheit?

7. A fluid exists at a temperature of 500 F abs. What temperature would a centigrade thermometer indicate when its bulb is inserted into the material?

8. A steam engine develops 100 hp continuously each hour. Find the heat equivalent of the work done in Btu per hour.

9. Steam flows in a pipe with a velocity of 200 fps. Find the heat equivalent of its kinetic energy per pound of steam.

10. Steam at 100 psia has 4.43 cu ft per lb and an enthalpy of 1187.2 Btu per lb. Find the heat equivalent of its flow energy. What is its internal energy per pound?

11. Steam at a pressure of 16 psia has a temperature of 216.3 F, a specific volume of 24.75 cu ft per lb, and an internal energy of 1078.7 Btu per lb. Calculate the enthalpy per pound of steam.

12. Forty pounds of dry air were raised in temperature from 32 to 212 F. Find the difference in the amounts of heat required if the process were at constant pressure instead of constant volume.

13. Eight thousand pounds of steel were cooled from 216 to 70 F. Find the amount of heat lost in the process.

14. A wall 12 in. thick has a surface area of 1200 sq ft and a thermal conductivity of 8. Calculate the amount of heat per hour which passes through the wall from one surface to the other when the inside-surface temperature is 68 F and the outside-surface temperature is 18 F.

15. A wall 8 in. thick has a surface area of 700 sq ft and transmits 80,000 Btu per hr by conduction when its inside-surface temperature is 65 F and its outside-surface temperature is −5 F. Find the thermal conductivity of the wall.

16. A wrought-iron pipe 450 ft long had a change of temperature from 65 to 230 F. Find the increase of the pipe length in inches.

17. A tank having a volume of 60 cu ft was filled with air having a temperature of 60 F and a pressure of 145 psig when the barometric pressure was 29.53 in. of mercury. Find the weight of air held in the tank.

18. Calculate the density of dry air which has a temperature of 75 F and is under a barometric pressure of 28 in. of mercury. Find the volume of one pound of the air under the conditions stated.

19. Find the pressure in pounds per square inch exerted by a water column 75 ft in height when the water temperature is 70 F.

20. Find the change of enthalpy necessary to convert 7500 lb of water at an initial temperature of 60 F into steam of 0.97 quality at a pressure of 150 psia.

21. Calculate the change of enthalpy necessary when 5000 lb of water at 180 F are converted into steam at 300 psia and a temperature of 500 F.

22. Calculate the total enthalpy of 20 lb of steam which exist at 150 psia and a total temperature of 450 F.

23. Calculate the volume of 700 lb of dry saturated steam which are under a pressure of 50 psia. Find the volume of 8000 lb of wet-saturated steam which has a quality of 0.98 and a pressure of 20 psia.

CHAPTER 2

PSYCHROMETRIC PROPERTIES OF AIR—HUMIDIFICATION AND DEHUMIDIFICATION

30. Air. Pure dry air is a mechanical mixture of the following gases: oxygen, nitrogen, carbon dioxide, hydrogen, argon, neon, krypton, helium, ozone, and xenon. The oxygen and the nitrogen make up the major portion of the combination. The other gases range from a few parts per 10,000 parts, by volume, to a mere trace in the case of xenon; their total never exceeds 100 parts per 10,000 parts (one per cent) of air. Therefore, dry air is generally considered as being made up in the following proportions: oxygen 20.91 per cent by volume and 23.15 per cent by weight, together with nitrogen 79.09 per cent by volume and 76.85 per cent by weight. Completely dry air does not exist in nature as water vapor in varying amounts is diffused through it. Air may be polluted by dust, dirt, bacteria, odor-bearing materials, and toxic gases of which carbon monoxide is an example.

Air, being one of the so-called perfect gases, approximately follows their laws, and the equations as stated in Art. 19 are applicable in many calculations where its weights and volumes are required at different pressures and temperatures. The volumes given at constant pressure and varying temperatures, in Table 8 were thus obtained.

The specific heats of air vary with its temperature conditions and also its state of maintenance, that is, either constant pressure or constant volume. For the temperature range of 32 to 400 F the mean specific heat of dry air at constant volume is 0.173. When constant-pressure conditions prevail the mean specific heat of dry air between 32 and 212 F is 0.2401, and between 32 and 392 F it is 0.2411. For ordinary computations the mean specific heat of dry air at constant pressure may be used as 0.24 with sufficient accuracy.

31. Mixtures of Air and Water Vapor. In psychrometric calculations the mixture considered is that of dry air and the water vapor (steam) mixed with it. The maximum amount of water vapor which can be mixed with dry air is dependent upon the absolute pressure and the temperature of the mixture. Air is saturated when it has diffused through it the maximum amount of water vapor (saturated steam in a stable state) that is possible under the conditions of temperature of the mixture and the absolute pressure of the dry air.

Saturated air exists at its dew-point temperature which is the one at which condensation of the water vapor begins and below which condensation of vapor from the mixture definitely takes place.

TABLE 8

Densities and Specific Volumes of Dry Air

Absolute pressure 29.92 in. of mercury

Temperature, Deg F	Weight per Cu Ft, Lb	Volume per Lb, Cu Ft	Temperature, Deg F	Weight per Cu Ft, Lb	Volume per Lb, Cu Ft
0	0.08633	11.58	180	0.06203	16.12
10	0.08449	11.83	200	0.06015	16.62
20	0.08273	12.08	220	0.05838	17.13
30	0.08104	12.34	240	0.05671	17.63
40	0.07942	12.59	260	0.05514	18.14
50	0.07785	12.84	280	0.05365	18.64
60	0.07636	13.09	300	0.05223	19.14
70	0.07492	13.34	350	0.04901	20.40
80	0.07353	13.59	400	0.04615	21.67
90	0.07219	13.85	450	0.04362	22.92
100	0.07090	14.10	500	0.04135	24.18
110	0.06966	14.35	550	0.03930	25.44
120	0.06845	14.60	600	0.03744	26.70
130	0.06729	14.85	700	0.03422	29.22
140	0.06617	15.11	800	0.03150	31.75
150	0.06509	15.36	900	0.02911	34.35
160	0.06403	15.61	1000	0.02718	36.79

When the temperature of an air-vapor mixture is held constant the amount of water vapor required to saturate one pound of dry air increases as the absolute pressure of the air is diminished and decreases as the pressure is increased. This fact is of importance at high altitudes where the barometric pressure is much lower than that existent at sea level. At a given pressure the temperature of the mixture has a very marked effect on the amount of water vapor that may be carried by a given quantity of air; the maximum amount increases as the air temperature rises and decreases as it falls. Under conditions of saturation the absolute partial pressure of the water vapor is that of saturated steam at the same temperature.

When air is not saturated at given conditions of pressure and temperature the vapor held is less than that possible at the temperature of the mixture; the vapor is superheated, and its partial pressure is less than that of saturated vapor at the same temperature. With conditions other than those of saturation the relative humidity, Art. 37, is less than 100 per cent, and the partial pressure of the vapor is that existent at the dew-point temperature of the mixture.

32. Weight of Water Vapor Required to Saturate Air. Considerable research has been done in connection with the psychrometric properties of moist air. The articles[1, 2] mentioned in the footnotes give data and indicate precise methods of calculation of the weight of water vapor necessary to saturate air. Neither air nor water vapor follows entirely the laws of perfect gases. Because of these deviations, temperature functions known as the second virial coefficients should be applied to the perfect-gas-law equation in the most exact calculations. For dry-air and water-vapor combinations interaction forces between the molecules should also be given consideration. However, at the temperatures and pressures involved in most air-conditioning problems the foregoing items need not be given attention, and the commonly used values based on computations including the use of Dalton's Law are entirely satisfactory and of sufficient accuracy.

According to Dalton's Law, air and water vapor each completely occupy a space filled by the mixture of the two as though the other were not present, and the total pressure of the mixture is the sum of the partial pressures of the water vapor and the dry air. Thus any mixture of water vapor and dry air, at any temperature, exists at an absolute pressure p which is the sum of p_1 the absolute partial pressure of the vapor and p_2 the absolute partial pressure of the dry air. Then $p = p_1 + p_2$ where each pressure is expressed in consistent units of either pounds per square inch or inches of mercury. Also, $p_2 = p - p_1$.

From Art. 19 the fundamental gas-law equation is $PV = WRT$ in connection with which in the following calculations the symbols p, v, and w will be used with the constant $R_1 = 85.71$ for water vapor and $R_2 = 53.35$ for dry air, the units of the constant being in each case foot-pounds per pound per degree Fahrenheit absolute temperature. According to Dalton's law the volume v_1 cu ft of water vapor at an absolute pressure of p_1 psi accruing from w_1 lb of moisture necessary to saturate one pound of dry air at an absolute temperature of T deg F is equal to the volume v_2 cu ft of dry air at an absolute pressure of p_2 psi at an absolute temperature of T deg F, or

$$v_1 = v_2 = \frac{w_1 R_1 T}{144 p_1} = \frac{1 \cdot R_2 \cdot T}{144(p - p_1)}$$

$$w_1 = \frac{1 \cdot R_2 p_1}{R_1(p - p_1)} = \frac{53.35 p_1}{85.71(p - p_1)} = 0.622 \frac{p_1}{(p - p_1)} \tag{18}$$

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[1] "Thermodynamic Properties of Moist Air," by J. A. Goff and S. Gratch, *ASHVE Trans.*, Vol. 51, 1945.

[2] "Low-Pressure Properties of Water in the Range −160 to 212 F, by J. A. Goff and S. Gratch, *ASHVE Journal Section of Heating, Piping and Air Conditioning*, February 1946.

Use of the foregoing equations may be made in building up data, Table 9, relative to the properties of both dry and saturated air. The following are used to illustrate the items involved.

Example. (*a*) Find the volume of one pound of dry air existing at a barometric pressure of 29.92 in. of mercury and a temperature of 68 F. (*b*) Find the volume of the mixture of one pound of dry air and the water vapor necessary to saturate it under the conditions of (*a*). (*c*) Calculate the weight of water vapor necessary to saturate one pound of dry air at the pressure and temperature given in (*a*). (*d*) Compute the density of the mixture.

Solution.

$$(a)\quad v = \frac{wR_2T}{144p} = \frac{1 \times 53.35(459.6 + 68)}{144 \times 29.92 \times 0.491} = 13.30 \text{ cu ft}$$

(*b*) $p_1 = 0.339$ psia (Table 6, Chap. 1) for steam at 68 F

$$p = 0.4912 \times 29.92 = 14.696 \text{ psia}$$

$$v_2 = \frac{w_2R_2T}{144(p - p_1)} = \frac{1 \times 53.35(459.6 + 68)}{144(14.696 - 0.339)} = 13.61 \text{ cu ft}$$

$$(c)\quad w_1 = 0.622\,\frac{p_1}{(p - p_1)} = \frac{0.622 \times 0.339}{(14.696 - 0.339)} = 0.01469 \text{ lb per lb of dry air}$$

or

$$0.01469 \times 7000 = 102.83 \text{ grains per pound of dry air}$$

$$(d)\quad d = \frac{1 + w_1}{v_2} = \frac{1 + 0.01469}{13.61} = 0.07455 \text{ lb per cu ft}$$

For temperatures below 32 F pressures and other data pertinent to properties of water vapor can be obtained from the *International Critical Tables* for use in calculating the psychrometric properties of air. The numerical effect of reduced air pressure is illustrated by the following problem case.

Example. A saturated air-vapor mixture exists at a barometric pressure of 28.32 in. of mercury and a temperature of 68 F. Find the volume of the mixture in cubic feet per pound of dry air involved and the weight of water vapor required to saturate one pound of the dry air.

Solution. The barometric pressure p is $0.491 \times 28.32 = 13.905$ psia, and the vapor pressure p_1 at 68 F is 0.339 psia. The volume of the mixture $v_2 = [1 \times 53.35 \times (459.6 + 68)]/[144(13.905 - 0.339)] = 14.409$ cu ft. The weight of water vapor to saturate one pound of dry air under the conditions stated is $w_1 = (0.622 \times 0.339)/(13.905 - 0.339) = 0.01554$ lb per lb or 108.78 grains per lb. The moisture to saturate w_1 may also be computed as $14.409/925.9 = 0.01556$ lb per lb where 925.9 is v_g the specific volume in cubic feet of one pound of saturated water vapor at a temperature of 68 F.

Comparison of the solutions of the two prior examples involving the calculation of the weight of water vapor required to saturate one pound of dry air indicates that for a given temperature (1) the vapor pressure p_1 for saturated water vapor remains constant irrespective of the barometric pressure p under which the mixture exists; (2) the partial pressure of the dry air p_2 decreases as the barometric pressure decreases; (3) the volume of the mixture, per pound of dry air and its contained water vapor, increases as the barometric pressure decreases; and (4) the weight of water vapor required to saturate one pound of dry air increases as the barometric pressure decreases. Later in this chapter it will become apparent that the greater amount of water vapor per pound of dry air the greater will be the total enthalpy of the mixture.

33. Humidity. The water vapor mixed with dry air is termed humidity. Special terms used in connection with the subject are absolute humidity and humidity ratio.

Absolute humidity is the actual weight of water vapor in grains (1 lb equals 7000 grains avoirdupois) or pounds per cubic foot of a mixture of air and water vapor.

Humidity ratio, often designated as specific humidity, is the weight, either in pounds or in grains, of water vapor diffused through one pound of dry air. The use of humidity ratios, rather than absolute humidities, is an aid in calculations although the results obtained are not entirely accurate. However, for nearly all calculations the errors involved by the use of humidity ratios may be ignored.

34. Enthalpy of Saturated Air. The heat held by a mixture of dry air and water vapor is generally reckoned from 0 F for the air and from 32 F for the vapor. The heat, Btu per lb of dry air, is $h_a = c_{pa}(t - 0)$ when c_{pa} is the mean specific heat of air at constant pressure and t is the dry-bulb temperature. The enthalpy of saturated water vapor is defined in Chap. 1 by equation 14, as $h_g = h_f + h_{fg}$ Btu per lb, and data may be obtained from Tables 5 and 6 of the same chapter. A convenient empirical expression for the enthalpy, in Btu per pound, of low-pressure, low-temperature saturated water vapor is

$$h_g = 1060.8 + 0.45t \tag{19}$$

where t is the temperature of the saturated mixture.

Example. Find the enthalpy of one pound of saturated water vapor at 70 F.

Solution. From Table 6, Chap. 1, $h_g = 1092.3$, or by equation 19, $1060.8 + (0.45 \times 70) = 1092.3$ Btu per lb.

The enthalpy of the mixture of one pound of dry air and the moisture to saturate it is

$$h_s = c_{pa}t + w_s h_g \tag{20}$$

where h_s = enthalpy of one pound of dry air and w_s the weight, in pounds per pound of dry air, of the moisture to saturate it, Btu.

t = temperature of the mixture, deg F.

h_g = enthalpy of saturated water vapor at the temperature t, Btu per lb.

Example. Find the enthalpy of one pound of dry air and the vapor to saturate it when the mixture exists under a barometric pressure of 29.92 in. of mercury and 68 F dry-bulb temperature.

Solution. By a prior calculation of part *c* in the example shown in Art. 32 the value of the weight of water vapor w_s for saturation is 0.01469 lb per lb. A calculated value of h_g is $1060.8 + (0.45 \times 68) = 1091.4$ Btu per lb of water vapor. The total enthalpy $h_s = (0.24 \times 68) + (0.01469 \times 1091.4) = 32.35$ Btu per lb.

35. Data for Saturated Air-Vapor Mixtures. The discussions of this chapter, thus far, have dealt with the properties of saturated air which change with variations of temperature and barometric pressure. Table 9 based on a common barometric pressure of 29.92 in. of mercury includes temperatures, corresponding vapor pressures, volumes, humidity ratios, and enthalpies. Other tables are in existence covering the same temperatures but with pressures greater or less than 29.92 in. of mercury. In such tables the vapor pressures are always the same at the same temperatures, but all other data are different than those for a given pressure.

36. Dry-Bulb and Wet-Bulb Temperatures. The dry-bulb temperature of an air-vapor mixture is the one indicated by any sort of a thermometer when its bulb, exposed to the atmosphere under consideration, is not in contact with water in either a solid or a liquid form. The foregoing statement is based on the premise that the reading of the thermometer is not affected by radiation of heat to or from the instrument.

The wet-bulb temperature of air (thermodynamic wet-bulb) for all practical purposes is the temperature of adiabatic saturation as defined later in Art. 43. It is the lowest temperature indicated by a moistened thermometer bulb when evaporation of the moisture takes place in a current of the air-vapor mixture as it moves rapidly over the instrument.

Dry- and wet-bulb temperatures are determined by the use of aspirating and sling psychrometers. Aspirating psychrometers usually have shields about the thermometer bulbs to prevent errors due to radiant energy and also some means of producing positive and rapid motion of the air-vapor mixture over the thermometer bulbs. For the best results the velocity of air motion should be from 1000 to 2000

TABLE 9

PROPERTIES OF MIXTURES OF AIR AND SATURATED WATER VAPOR* †

Barometric pressure 29.92 in. of mercury

Air Temperature, Deg F	Pressure of Saturated Vapor, Psia	Volume		Weight of Water Vapor for Saturation of One Pound of Dry Air		Total Enthalpy or Total Heat of		
		One Pound of Dry Air, Cu Ft	One Pound of Dry Air and Vapor to Saturate, Cu Ft	Grains	Pounds	One Pound of Dry Air above 0 F, Btu	Water Vapor to Saturate One Pound Of Dry Air, Btu above 32 F	One Pound of Dry Air Saturated, Btu
0	0.0185	11.58	11.59	5.50	0.000785	0.00	0.8317	0.8317
5	0.0240	11.70	11.72	7.12	0.001017	1.200	1.080	2.280
10	0.0309	11.83	11.85	9.18	0.001311	2.400	1.395	3.795
15	0.03963	11.96	11.99	11.77	0.001682	3.599	1.793	5.392
20	0.0539	12.09	12.13	16.03	0.002290	4.802	2.447	7.249
21	0.0562	12.11	12.16	16.73	0.002390	5.042	2.555	7.597
22	0.0587	12.14	12.19	17.46	0.002494	5.282	2.667	7.949
23	0.0612	12.16	12.21	18.21	0.002601	5.522	2.783	8.305
24	0.0638	12.19	12.24	18.99	0.002713	5.762	2.903	8.665
25	0.0665	12.21	12.27	19.80	0.002829	6.003	3.028	9.031
26	0.0693	12.24	12.30	20.64	0.002949	6.243	3.158	9.401
27	0.0722	12.26	12.33	21.51	0.003073	6.483	3.293	9.776
28	0.0752	12.29	12.35	22.42	0.003203	6.723	3.433	10.156
29	0.0783	12.31	12.38	23.37	0.003338	6.963	3.579	10.542
30	0.0816	12.34	12.41	24.35	0.003479	7.203	3.731	10.934
31	0.0850	12.36	12.44	25.37	0.003624	7.443	3.889	11.332
32	0.0886	12.39	12.47	26.43	0.003776	7.683	4.053	11.736
33	0.0923	12.42	12.49	27.53	0.003933	7.923	4.223	12.146
34	0.0961	12.44	12.52	28.67	0.004096	8.163	4.400	12.563
35	0.1000	12.47	12.55	29.85	0.004264	8.404	4.583	12.987
36	0.1041	12.49	12.58	31.07	0.004439	8.644	4.773	13.417

37	0.1083	12.52	12.61	32.33	0.004619	8.884	4.970	13.854
38	0.1126	12.54	12.64	33.64	0.004806	9.124	5.174	14.298
39	0.1171	12.57	12.67	35.00	0.00500	9.364	5.385	14.749
40	0.1217	12.59	12.70	36.41	0.005201	9.604	5.603	15.207
41	0.1265	12.62	12.73	37.87	0.005410	9.844	5.829	15.673
42	0.1315	12.64	12.76	39.38	0.005626	10.084	6.062	16.146
43	0.1367	12.67	12.79	40.94	0.005849	10.324	6.303	16.627
44	0.1420	12.69	12.82	42.55	0.006079	10.564	6.553	17.117
45	0.1475	12.72	12.85	44.21	0.006314	10.805	6.812	17.617
46	0.1532	12.74	12.88	45.93	0.006561	11.045	7.081	18.126
47	0.1591	12.77	12.91	47.71	0.006816	11.285	7.359	18.644
48	0.1652	12.79	12.94	49.55	0.007079	11.525	7.647	19.172
49	0.1715	12.82	12.97	51.45	0.007350	11.765	7.946	19.711
50	0.1780	12.84	13.00	53.42	0.007632	12.005	8.256	20.261
51	0.1848	12.87	13.03	55.46	0.007923	12.245	8.577	20.822
52	0.1918	12.89	13.06	57.57	0.008224	12.485	8.909	21.394
53	0.1989	12.92	13.10	59.75	0.008536	12.725	9.252	21.977
54	0.2063	12.95	13.13	62.01	0.008859	12.965	9.606	22.571
55	0.2140	12.97	13.16	64.35	0.009193	13.206	9.972	23.178
56	0.2219	13.00	13.19	66.77	0.009539	13.446	10.350	23.796
57	0.2300	13.02	13.23	69.27	0.009896	13.686	10.740	24.426
58	0.2384	13.05	13.26	71.85	0.01026	13.926	11.143	25.069
59	0.2471	13.07	13.29	74.52	0.01065	14.166	11.560	25.726
60	0.2561	13.10	13.33	77.28	0.01104	14.406	11.991	26.397
61	0.2654	13.12	13.36	80.13	0.01145	14.646	12.437	27.083
62	0.2749	13.15	13.40	83.07	0.01187	14.886	12.899	27.785
63	0.2848	13.17	13.43	86.11	0.01230	15.126	13.376	28.502
64	0.2949	13.20	13.47	89.25	0.01275	15.366	13.869	29.235
65	0.3054	13.22	13.50	92.49	0.01321	15.607	14.379	29.986
66	0.3162	13.25	13.54	95.83	0.01369	15.847	14.905	30.752
67	0.3273	13.27	13.57	99.28	0.01418	16.087	15.448	31.535
68	0.3388	13.30	13.61	102.84	0.01469	16.327	16.009	32.336
69	0.3506	13.32	13.65	106.51	0.01522	16.567	16.589	33.156
70	0.3628	13.35	13.69	110.30	0.01576	16.807	17.187	33.994
71	0.3754	13.37	13.72	114.21	0.01632	17.047	17.804	34.851
72	0.3883	13.40	13.76	118.25	0.01689	17.287	18.441	35.728
73	0.4016	13.42	13.80	122.42	0.01749	17.527	19.099	36.626

TABLE 9 (*Continued*)

Air Temperature, Deg F	Pressure of Saturated Vapor, Psia	Volume		Weight of Water Vapor for Saturation of One Pound of Dry Air		Total Enthalpy or Total Heat of		
		One Pound of Dry Air, Cu Ft	One Pound of Dry Air and Vapor to Saturate, Cu Ft	Grains	Pounds	One Pound of Dry Air above 0 F, Btu	Water Vapor to Saturate One Pound of Dry Air, Btu above 32 F	One Pound of Dry Air Saturated, Btu
74	0.4153	13.45	13.84	126.72	0.01810	17.767	19.778	37.545
75	0.4294	13.47	13.88	131.16	0.01874	18.008	20.479	38.487
76	0.4440	13.50	13.92	135.75	0.01939	18.248	21.203	39.451
77	0.4590	13.52	13.96	140.48	0.02007	18.488	21.951	40.439
78	0.4744	13.55	14.00	145.36	0.02077	18.728	22.724	41.452
79	0.4903	13.58	14.04	150.40	0.02149	18.968	23.522	42.490
80	0.5067	13.60	14.09	155.61	0.02223	19.208	24.346	43.554
81	0.5236	13.63	14.13	160.98	0.02299	19.45	25.20	44.65
82	0.5409	13.65	14.17	166.52	0.02379	19.69	26.07	45.76
83	0.5588	13.68	14.22	172.23	0.02460	19.93	26.98	46.91
84	0.5772	13.70	14.26	178.11	0.02544	20.17	27.91	48.08
85	0.5960	13.73	14.31	184.17	0.02631	20.41	28.87	49.28
86	0.6153	13.75	14.35	190.41	0.02720	20.65	29.86	50.51
87	0.6352	13.78	14.40	196.83	0.02812	20.89	30.88	51.77
88	0.6556	13.80	14.45	203.44	0.02906	21.13	31.94	53.07
89	0.6765	13.83	14.49	210.25	0.03004	21.37	33.02	54.39
90	0.6980	13.85	14.54	217.27	0.03104	21.61	34.14	55.75
91	0.7201	13.88	14.59	224.51	0.03207	21.85	35.29	57.14
92	0.7429	13.90	14.64	231.98	0.03314	22.09	36.48	58.57
93	0.7662	13.93	14.69	239.68	0.03424	22.33	37.71	60.04
94	0.7902	13.95	14.75	247.62	0.03532	22.57	38.97	61.54
95	0.8149	13.98	14.80	255.81	0.03654	22.81	40.27	63.08
96	0.8403	14.00	14.85	264.25	0.03775	23.05	41.62	64.67
97	0.8663	14.03	14.91	272.95	0.03899	23.29	43.00	66.29
98	0.8930	14.05	14.96	281.92	0.04027	23.53	44.43	67.96

99	0.9205	14.08	15.02	291.17	0.04160	23.77	45.91	69.68
100	0.9487	14.10	15.08	300.70	0.04296	24.01	47.43	71.44
101	0.9776	14.13	15.14	310.52	0.04436	24.25	49.00	73.25
102	1.0072	14.15	15.20	320.64	0.04581	24.49	50.61	75.10
103	1.0377	14.18	15.26	331.1	0.0473	24.73	52.28	77.01
104	1.0689	14.21	15.32	341.8	0.0488	24.97	54.00	78.97
105	1.1009	14.23	15.38	352.9	0.0504	25.21	55.78	80.99
106	1.1338	14.26	15.45	364.3	0.0520	25.45	57.61	83.06
107	1.1675	14.28	15.51	376.0	0.0537	25.69	59.49	85.18
108	1.2020	14.31	15.58	388.2	0.0555	25.93	61.44	87.37
109	1.2374	14.33	15.65	400.7	0.0572	26.17	63.44	89.61
110	1.274	14.36	15.72	413.6	0.0591	26.41	65.51	91.92
111	1.311	14.38	15.79	426.8	0.0610	26.65	67.64	94.29
112	1.349	14.41	15.86	440.6	0.0629	26.89	69.84	96.73
113	1.388	14.43	15.94	454.7	0.0650	27.13	72.10	99.23
114	1.429	14.46	16.01	469.3	0.0670	27.37	74.44	101.81
115	1.470	14.48	16.09	484.3	0.0692	27.61	76.85	104.46
116	1.512	14.51	16.17	499.9	0.0714	27.85	79.44	107.92
117	1.555	14.53	16.25	515.9	0.0737	28.09	82.01	110.10
118	1.599	14.56	16.34	532.4	0.0761	28.33	84.66	112.99
119	1.645	14.58	16.42	549.5	0.0785	28.57	87.39	115.96
120	1.691	14.61	16.51	567.1	0.0810	28.81	90.21	119.02
121	1.739	14.63	16.60	585.2	0.0836	29.05	93.12	122.17
122	1.788	14.66	16.69	604.0	0.0863	29.29	96.13	125.42
123	1.838	14.68	16.78	623.3	0.0890	29.53	99.23	128.76
124	1.889	14.71	16.88	643.2	0.0919	29.77	102.44	132.21
125	1.941	14.73	16.98	663.8	0.0948	30.01	105.75	135.76
126	1.995	14.76	17.08	685.1	0.0979	30.25	109.17	139.42
127	2.049	14.78	17.18	707.0	0.1010	30.49	112.71	143.20
128	2.105	14.81	17.29	729.7	0.1042	30.73	116.37	147.10
129	2.163	14.84	17.39	753.1	0.1076	30.97	120.15	151.12
130	2.221	14.86	17.51	777.2	0.1110	31.21	124.05	155.26

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* Reproduced by permission of the General Electric Company.

† Values for 0 to 15 F are from data prepared by Professor W. M. Sawdon for the *ASHVE* Guide, 1938.

fpm. The whirling psychrometer, illustrated by Fig. 4 consists of two accurate mercurial thermometers mounted upon a frame which has a handle that permits the assembly to be rapidly rotated to produce the necessary air motion. One thermometer is fixed with its bulb projecting beyond the bulb of the other. This projecting bulb is covered with a very light fabric gauze which is moistened with pure water held at a temperature near to that to be determined. The other bulb is kept absolutely dry. After rapid rotation of the thermometers, to give the necessary air motion, readings are taken of both thermometers when the minimum wet-bulb temperature is indicated. The lower temperature given by the wet-bulb thermometer, in partially saturated air, is due to the evaporation of moisture from the gauze and the transformation of sensible into latent heat. When the air is saturated there is no evaporation at the moistened bulb, and the two thermometers read alike. At any given dry-bulb temperature the greater the depression of the wet-bulb temperature reading below that of the dry-bulb thermometer the smaller is the amount of water vapor held in the mixture.

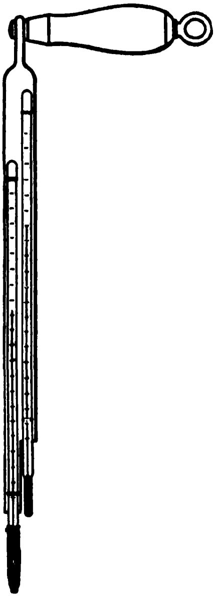

FIG. 4. Sling psychrometer.

Hygrometers which have their wet bulbs standing in still air may give very erroneous results. The accurate determination of the wet-bulb temperature is very important as at any given dry-bulb temperature it is indicative of the amount of moisture held in the air and the enthalpy of a unit quantity of the mixture of air and water vapor.

37. Relative Humidity and Per Cent of Saturation. The amount of water vapor diffused through dry air in an air-vapor mixture may vary from practically nothing to that necessary for saturation conditions. As an indication of the degree of saturation of the mixture the terms relative humidity and per cent of saturation are employed. These terms are not identical; one cannot be used exactly for the other as they are not interchangeable.

Relative humidity, usually expressed as a percentage, is preferably defined as the ratio of the actual partial pressure of water vapor of the air-vapor mixture to the partial pressure normal for saturated water vapor at the existent dry-bulb temperature of the mixture. *Per cent of saturation* is 100 times the ratio of the weight of water vapor actually held per pound of dry air to that necessary to saturate the pound of dry air at the same dry-bulb temperature and absolute total pressure.

38. The Calculation of Vapor Pressures and Relative Humidities. The vapor pressure of an air-vapor mixture is always the pressure corresponding to that of saturated water vapor existing at its dew-point temperature. Since the barometric pressure and the dry- and wet-bulb temperatures are readily obtainable, vapor pressures may be secured by calculations from measured data. Equation 21 which follows was developed by Dr. Willis H. Carrier and affords a satisfactory means of making such computations. In the equation all pressures are expressed consistently either as absolute units in pounds per square inch or inches of mercury.

$$p_1 = p' - \frac{(p - p')(t - t')}{(2831 - 1.43t')} \tag{21}$$

where p = barometric pressure.
p_1 = actual vapor pressure.
p' = saturation pressure at the wet-bulb temperature.
t = dry-bulb temperature, deg F.
t' = wet-bulb temperature, deg F.

A development of equation 21 may be made by using, with proper substitutions, equations 18 and 31, Art. 43, as equalities. When an expression of an actual partial or vapor pressure is made in this manner, one portion of the derived equation is of such small magnitude that it may be safely ignored. Only those portions shown in equation 21 are necessary for the calculation of results that are as accurate as the data obtained by the use of an ordinary psychrometer. The constants as used in equation 21 differ by small amounts from those first given by Dr. Willis H. Carrier. The reason is that a quantity of 0.45 Btu per lb of either saturated or superheated vapor is taken as the enthalpy change per deg F of temperature change.

The relative humidity definition of Art 37, expressed in equation form, is

$$\phi = \frac{p_1}{p_s} 100 \tag{22}$$

where p_1 = the actual vapor pressure of the mixture.
p_s = the pressure of saturated water vapor at the dry-bulb temperature.

Likewise an expression for per cent saturation is

$$s = \frac{w_1}{w_s} 100 \tag{23}$$

in which w_1 is the weight of water vapor in pounds per pound of dry air and w_s is the amount of water vapor required to saturate one pound of dry air at the dry-bulb temperature (dbt) of the mixture.

Example. Find the relative humidity and the per cent of saturation for an air-vapor mixture existing under the conditions of a barometric pressure of 29.92 in. of mercury (14.696 psia), 80 F dbt and 60 F wbt.

Solution. Table 6, Chap. 1, indicates that at 60 F wbt the saturation pressure p' for water vapor is 0.2563 psia, and p_s at 80 F dbt is 0.5069 psia. Then by equation 21 the actual vapor pressure is found to be $p_1 = 0.2563 - [(14.696 - 0.2563)(80 - 60)]/[2831 - (1.43 \times 60)] = 0.2563 - 0.1052 = 0.1511$ psia. The relative humidity is $\phi = (0.1511 \times 100)/0.5069 = 29.8$ per cent. The actual weight of water vapor required to saturate one pound of dry air, when the vapor pressure is 0.1511 psia and the corresponding dew-point temperature is 45.6 F, according to equation 18 is $w_1 = (0.622 \times 0.1511)/(14.696 - 0.1511) = 0.00646$ lb per lb. The weight of water vapor required to saturate one pound of dry air at 80 F dbt is $w_s = (0.622 \times 0.5069)/(14.696 - 0.5069) = 0.0222$ lb per lb. Thence by definition the per cent of saturation is $s = (0.00646 \times 100)/0.0222 = 29.1$ per cent.

The relative humidity as thus determined is not identical with the per cent of saturation. Relative humidity can be applied only to the vapor pressure in the calculation of humidity ratio (specific humidity) while the per cent of saturation is used to find the actual humidity ratio by applying it to the weight of water vapor required to saturate one pound of dry air at the dry-bulb temperature.

In the foregoing and similar problems use is made of the actual partial pressure p_1 to find the dew-point temperatures. The dew-point t_d exists when the humidity ratio w_1 (specific humidity) is sufficient to saturate one pound of dry air when the vapor pressure is the actual one of p_1 psia. Either Table 6, Chap. 1, or Table 9 may be used to find the dew-point temperature for any vapor pressure.

39. Enthalpy of Air-Vapor Mixtures for Any Condition. The enthalpy h of a pound of dry air and its contained moisture is often expressed as

$$h = c_{pa}t + w[h_{fg}' + c_{ps}(t - t')] \tag{24}$$

where c_{pa} = mean specific heat of air at constant pressure, 0.24.

t = air dry-bulb temperature, deg F.

t' = air wet-bulb temperature, deg F.

w = weight of moisture per pound of dry air, lb.

h_{fg}' = enthalpy of evaporation at the wet-bulb temperature, Btu per lb.

c_{ps} = mean specific heat of the vapor, 0.45.

The proof offered in Art. 43, in connection with an adiabatic process, results in equation 30 which can be arranged as

$$c_{pa}t_2 + w_2h_{fg,2} = c_{pa}t_1 + w_1[h_{fg,2} + c_{ps}(t_1 - t_2)]$$

The enthalpy represented by the right-hand side of the foregoing statement is the same as that of equation 24 when $t = t_1$, $t' = t_2$, $w = w_1$, and $h_{fg}' = h_{fg,2}$.

Equation 24 does not include the enthalpy of the liquid h_f' at the wet-bulb temperature; only enthalpy of evaporation of dry saturated vapor at the wet-bulb temperature and the heat of superheat above this temperature are involved. If the enthalpy of the liquid h_f' were included the equation could be written

$$h = c_{pa}t + w[h_f' + h_{fg}' + c_{ps}(t - t')] \tag{25}$$

For practical purposes the total enthalpy of a mixture of dry air and water vapor may be considered as being the same for all conditions of constant pressure where the wet-bulb temperature does not change as the dry-bulb temperatures and the humidity ratios of the mixture vary. This assumption involves some inaccuracy. For saturation conditions with dry- and wet-bulb temperatures t_s and t_s' equal and with a humidity ratio w_s, the total enthalpy h_s is not exactly equal to the total enthalpy h_1 for an unsaturated mixture having a wet-bulb temperature t_1' equal to t_s' and a humidity ratio w_1 together with a dry-bulb temperature t_1. The correct value of h_s is

$$h_s = h_1 + (w_s - w_1)h_f' \tag{26}$$

where h_s = total enthalpy for a saturated air-vapor mixture at a wet-bulb temperature of t_s', deg F, Btu per lb of dry air and contained water vapor.

h_1 = total enthalpy of an unsaturated air-water-vapor mixture having a wet-bulb temperature $t_1' = t_s'$, deg F, Btu per lb of dry air and contained water vapor.

w_s = weight of water vapor lb per lb of dry air at wet-bulb temperature $t_s' = t_1'$ and dry-bulb temperature t_s, deg F.

w_1 = weight of water vapor lb per lb of dry air at wet-bulb temperature $t_1' = t_s'$ and dry-bulb temperature t_1, deg F.

h_f' = enthalpy of the liquid at the wet-bulb temperature $t_s' = t_1'$, Btu per lb.

When h_s is known equation 26 can be used to determine h_1.

Many engineers when making air-conditioning calculations prefer to work with a quantity known as the *sigma function,* which is equal to

$$h_{sf} = c_{pa}t + w_1h_{fg}' \tag{27}$$

where h_{sf} = sigma function, Btu per lb of dry air and contained water vapor.
c_{pa} = mean specific heat of air at constant pressure, 0.2401.
t = dry-bulb temperature of air-vapor mixture, deg F.
w_1 = weight of water vapor per lb of dry air, lb.
h_{fg}' = enthalpy of evaporation at the wet-bulb temperature of the mixture, Btu per lb.

Use of the sigma function is made to eliminate the variation of the enthalpy at constant wet-bulb temperature when the enthalpy of the liquid is involved. A correction may be made after the use of values of the sigma function to offset this variation.

Attention is called to the fact that in equation 25 each degree of superheat of the vapor represents 0.45 Btu per lb and that in equation 19, for saturated vapor, a change of one degree Fahrenheit temperature of the fluid represents the same amount of heat, that is, Btu per lb of vapor. Because of this coincidence the expression

$$h = c_{pa}t + wh_g \tag{28}$$

may be used to calculate the enthalpy of any mixture of dry air and water vapor.

where h = total enthalpy, Btu per lb of dry air and its contained water vapor.
c_{pa} = mean specific heat of dry air at constant pressure, 0.2401.
t = dry-bulb temperature of the mixture, deg F.
w = humidity ratio, weight of water vapor, lb per lb of dry air.
h_g = enthalpy of saturated water vapor at the dry-bulb temperature t, Btu per lb.

Example. Find (*a*) the total enthalpy of one pound of dry air and the moisture which it carries for the following conditions: barometric pressure, 14.696 psia; 80 F dbt; and 60 F wbt; (*b*) the total enthalpy of one pound of dry air when saturated at 60 F wbt when the barometric pressure is 14.696 psia. (*c*) By use of equation 26 correct the result of *a* to the conditions of *b*.

Solution. (*a*) The solution of the problem of Art. 38 for the same conditions gave the humidity ratio w equal to 0.00646 lb per lb. Then, by equation 25, $h = (0.2401 \times 80) + 0.00646[28.06 + 1059.9 + 0.45(80 - 60)] = 26.294$ Btu, and, by equation 28, $h = (0.2401 \times 80) + (0.00646 \times 1096.6) = 26.292$ Btu. These computed numerical values are about 0.39 of one per cent less than the 26.397 Btu indicated in Table 9 for air saturated at the wet-bulb temperature of 60 F. (*b*) The moisture or water vapor required to saturate one pound of dry air at 60 F is $w_s = (0.622 \times 0.2563)/(14.696 - 0.2563) = 0.01105$ lb per lb. $h_s = (0.2401$

× 60) + (0.01105 × 1088) = 26.428 Btu, (c) by equation 26, h_s = 26.292 + (0.01105 − 0.00646)28.06 = 26.421 Btu.

40. Psychrometric Charts. These are diagrams which show graphically properties of air-vapor mixtures for conditions of saturation and otherwise are of great convenience in solving problems of air conditioning. These diagrams are constructed in the form of Mollier[3] and psychrometric charts of different forms. Each has its advantages. One form of psychrometric chart, as devised by the General Electric Company and reproduced by permission, is illustrated by Fig. 5. In the construction of this particular chart dry-bulb temperatures serve as abscissas and weights of water vapor per pound of dry air along with the corresponding vapor pressures as ordinates. Inasmuch as relative humidities represent vapor-pressure ratios, curves representing these are placed on the diagram with dry-bulb temperatures as abscissas and actual vapor pressures together with the corresponding humidity ratios as ordinates. At saturation conditions (100 per cent relative humidity) the air dry- and wet-bulb temperatures and the dew-point temperature are identical for a definite mixture. Consequently these values are placed on the saturation curve directly above the dry-bulb temperatures. Lines representing constant wet-bulb temperatures are drawn diagonally from the saturation curve to fit conditions of higher dry-bulb temperatures and lesser amounts of moisture per pound of dry air. Although a line of constant wet-bulb temperature does not exactly represent a line of constant total enthalpy (also designated as total heat), Art. 39, on this particular chart they are so used for one pound of dry air and its vapor content at any dry-bulb temperature. Therefore, lines of constant total heat or enthalpy are along or parallel to the wet-bulb temperature lines. Vapor pressures for any conditions shown on the chart are indicated at its left side, and a value can be obtained by passing horizontally from the point representing any condition to the vapor-pressure scale. Volumes of one pound of dry air and its contained water vapor also appear on the chart.

41. Use of the Psychrometric Chart. The following examples serve to indicate the use of the psychrometric chart.

Example. An air-vapor mixture has 70 F dbt and 61 F wbt when the barometric pressure is 29.92 in. of mercury. From the psychrometric chart of Fig. 5 find the volume of one pound of dry air and the moisture to saturate it, the relative humidity, the humidity ratio, the dew-point temperature, the vapor pressure, and the total heat or enthalpy of the mixture.

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[3] "Mollier Diagram for Moist Air," by John A. Goff, *ASHVE Guide*, 1948.

Solution. The intersection of the vertical line through 70 F dbt and the diagonal line representing 61 F wbt fixes 13.54 cu ft as the volume of the mixture per pound of dry air and its contained water vapor and also 60 per cent as the relative humidity for the stated conditions. Passing from the point of intersection to the right the humidity ratio or specific humidity is found to be 65.5 grains of moisture per pound of dry air. Moving horizontally to the left from the point of intersection the dew-point temperature is read at the saturation curve (100 per cent relative humidity) as 55.5 F, and the vapor-pressure scale at the edge of the chart shows 0.215 psia. The total heat or enthalpy is found by following the 61 F wbt line through the point of intersection to the total heat or enthalpy scale where 27.10 Btu per lb of dry air are indicated.

Whenever the barometric pressure does not differ greatly from 29.92 in. of mercury, data taken from a chart drawn for such a pressure may be used with small significant errors. However, when the barometric pressure is low the errors accruing from the use of a standard chart may amount to more than those ordinarily permissible. The corrections of some of the data taken from a standard chart may be easily made by the simple application of pressure ratios, but for others the problem becomes more complicated. When such a situation arises the psychrometric data can be computed almost as readily as a chart can be read and corrections applied to the data thus secured.

42. Actual Psychrometric Data vs. Data from a Standard Chart. The following example with its tabular data will show the difference between chart and actual values when the barometric pressures are not the same.

Example. Find by calculation the psychrometric properties of an air-vapor mixture for the temperatures stated in the example of Art. 41 when the barometric pressure is 28 in. of mercury. Tabulate and compare the data thus obtained with those secured by the method of Art. 41.

Solution. From the equations of prior articles the following may be found: barometric pressure $p = 0.491 \times 28 = 13.748$ psia; vapor pressure $p_1 = 0.2655 - [(13.748 - 0.2655)(70 - 61)]/[2831 - (1.43 \times 61)] = 0.221$ psia; volume of the mixture $v_2 = 1 \times 53.35(459.6 + 70)/144(13.748 - 0.221) = 14.50$ cu ft; the

Psychrometric Properties of an Air-Vapor Mixture

Item	By Standard Chart	By Calculation
Barometer, in. of mercury	29.92	28.00
Dry-bulb, deg F	70	70
Wet-bulb, deg F	61	61
Vapor pressure, psia	0.215	0.221
Volume, cu ft per lb	13.54	14.50
Relative humidity, per cent	60	60.8
Dew point, deg F	55.5	55.9
Humidity ratio, grains	65.5	71.12
Enthalpy, Btu per lb	27.10	27.904

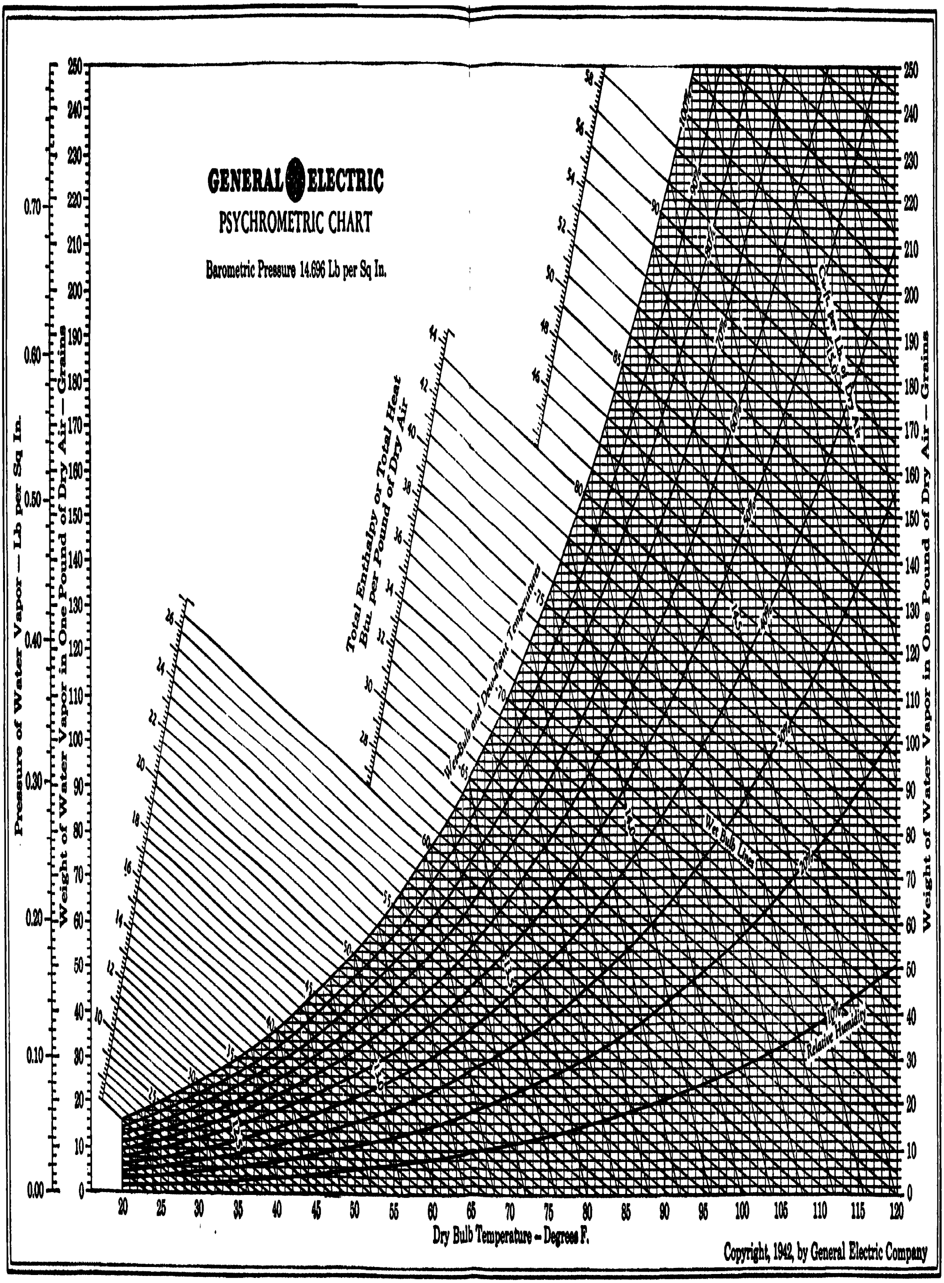

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Fig. 5. Psychrometric chart.

vapor pressure p_1 from Table 6, Chap. 1, is 0.3631 psia at 70 F; the relative humidity $\phi = (0.221 \times 100)/0.3631 = 60.8$ per cent; the dew-point temperature is found from either Table 6, Chap. 1, or Table 9 as 55.9 F which is the saturation temperature of water vapor corresponding to an absolute pressure of 0.221 psi; the humidity ratio $w_1 = (0.622 \times 0.221)/(13.748 - 0.221) = 0.01016$ lb per lb or 71.12 grains; and the total enthalpy $h = (0.2401 \times 70) + (0.01016 \times 1092.3) = 27.904$ Btu per lb of dry air.

43. Adiabatic Saturation of Air. This fundamental term[4] was defined by Dr. Willis H. Carrier and is applicable, under certain conditions, in the humidification of air with an accompanying reduction of its dry-bulb temperature and in the drying of materials by the use of heated air. Air may be either completely or partially adiabatically saturated with vapors from water and other liquids. In this discussion water vapor only will be considered.

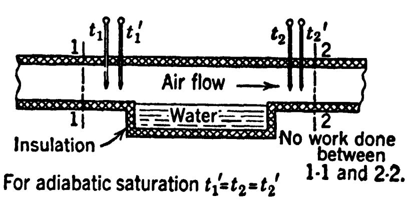
t_1
t_1'
t_2
t_2'
1
1
2
2
Air flow
Water
Insulation
No work done between 1-1 and 2-2.
For adiabatic saturation $t_1' = t_2 = t_2'$

FIG. 6. Apparatus for adiabatic saturation of air with water vapor.

Experimentally the process can be carried out in a well-insulated chamber, Fig. 6, where heat is neither received from nor given to outside bodies (adiabatic conditions) and where work is not done upon the air as it passes through the apparatus under a condition of a steady flow. The moving air stream is brought into intimate contact with a large surface area of a sufficient quantity of water held within the chamber. The process is one involving a condition of constant enthalpy with sensible heat being transformed to latent heat. For the purposes of simplicity the heat equivalent of any work done upon the water by a pump used to spray the water and thereby increase its surfaces in contact with the air will be ignored at this time.

When constant and normal conditions of operation have been established, theoretically the water temperature and the final dry- and wet-bulb temperatures of the saturated air-vapor mixture leaving the apparatus are the same as the wet-bulb temperature of the entering air or air-vapor mixture.

Under the conditions just specified the process of evaporating water in air to saturate it, by having a part of the sensible heat of the entering air transformed into latent heat, is that of *adiabatic saturation.*

For air initially dry the heat transformation is expressed in equation form as

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[4] Rational Psychrometric Formulae," by Willis H. Carrier, *ASME Trans.*, Vol. 33, 1911.

$$h_{fg}'w' = c_{pa}(t_1 - t') \qquad (29)$$

where t_1 = the initial dry-bulb temperature of the air, deg F.
t' = the final dry-bulb, the final wet-bulb, and the initial wet-bulb temperature of the air, deg F.
c_{pa} = mean specific heat of air at constant pressure, 0.24.
w' = weight of moisture per pound of dry air leaving the apparatus saturated at the wet-bulb temperature t'. lb.
h_{fg}' = enthalpy or latent heat of evaporation at the wet-bulb temperature t', Btu per lb of vapor added.

A discussion of the enthalpy of an air-vapor mixture for any condition was included in Art. 39. Attention was called to the slight variation of the total enthalpy for constant wet-bulb temperature, and equation 26 was given as a means of calculating a value of h, for saturation conditions when the initial state is not that of saturation. In the following discussion of adiabatic saturation of air, which is not initially dry, the subscripts 1 and 2 will be used to designate the range of conditions. The total enthalpy of one pound of dry saturated water vapor will be computed as $1060.8 + 0.45t$ and the enthalpy of one pound of the liquid at the evaporation temperature as $h_f = (t - 32)$. The following energy balance may be written for air not initially dry:

$$c_{pa}t_1 + w_1(1060.8 + 0.45t_1) - w_1(t_2 - 32) = c_{pa}t_2 + w_2(1060.8 + 0.45t_2) - w_2(t_2 - 32)$$

$$c_{pa}t_1 + w_1(1060.8 + 0.45t_1) + (w_2 - w_1)(t_2 - 32) = c_{pa}t_2 + w_2(1060.8 + 0.45t_2)$$

In the foregoing equations $h_{f,2}$ is taken as equal to $(t_2 - 32)$, and $w_1(1060.8 + 045t_1) = w_1[1060.8 + 0.45t_2 + c_{ps}(t_1 - t_2)]$. Also $h_{fg,2} = (1060.8 + 0.45t_2) - h_{f,2}$. With these substitutions

$$(w_2 - w_1)h_{fg,2} = (c_{pa} + c_{ps}w_1)(t_1 - t_2) \qquad (30)$$

where $h_{fg,2}$ = enthalpy of evaporation at temperature t_2, Btu per lb. Theoretically $t_1' = t_2' = t_2$ where t_1' and t_2' are the initial and the final wet-bulb temperatures, and t_2 is the final dry-bulb temperature, deg F.
w_2 = final weight of water vapor per pound of dry air, lb.
w_1 = initial weight of water vapor per pound of dry air, lb.
c_{pa} = mean specific heat of air at constant pressure, 0.24.
c_{ps} = mean specific heat of the vapor, 0.45.
t_1 = initial dry-bulb temperature of the air, deg F.
t_2 = final dry-bulb temperature of the air, deg F. t_2 is equal to t' the wet-bulb temperature when the air is saturated.

Equation 30 may be utilized to determine the amount of moisture initially held by the air when the final weight of moisture per pound of dry air is known. Equation 30 in rearranged form is

$$w_1 = \frac{w_2 h_{fg,2} - c_{pa}(t_1 - t_2)}{h_{fg,2} + c_{ps}(t_1 - t_2)} \tag{31}$$

44. Humidifying Efficiency. Complete adiabatic saturation is not always possible with commercial equipment such as air washers. Therefore a term known as humidifying efficiency is used to express the percentage of reduction of the dry-bulb temperature actually obtained in such equipment to that theoretically possible with complete adiabatic saturation of air. The maximum possible reduction of the dry-bulb temperature of air by the process of adiabatic saturation is equal to the initial wet-bulb temperature depression below that of the dry-bulb temperature of the entering air. Sometimes the expression 1 minus the humidifying efficiency, expressed decimally, is referred to as the *by-pass factor.* Humidifying efficiency is stated in equation form as

$$e_h = \frac{t_1 - t_2}{t_1 - t_1'} 100 \tag{32}$$

where e_h = humidifying efficiency, per cent.
t_1 = initial dry-bulb air temperature, deg F.
t_2 = final air dry-bulb temperature, deg F.
t_1' = initial air wet-bulb temperature, deg. F

Example. Air, having dry- and wet-bulb temperatures of 95 and 75 F, is passed through a spray using recirculated spray water which has acquired a constant operating temperature. Find the humidifying efficiency of the apparatus when the final dry-bulb temperature of the air is 77 F.

Solution.

$$e_h = \frac{95 - 77}{95 - 75} 100 = 90 \text{ per cent}$$

45. Evaporative Cooling. When conditions are favorable the partial adiabatic saturation of air can be used to reduce its dry-bulb temperature in hot weather. As moisture must be added to the air in evaporative cooling the lowest dry-bulb temperature possible is that equal to the wet-bulb temperature of the entering air. Consequently, when the initial relative humidity is high the reduction of dry-bulb temperature possible is less than when the air is initially drier. The addition of moisture to the air, in the reduction of its dry-bulb temperature, may render the air condition decidedly uncomfortable for human occupants of a space where cooling of the air is attempted by such a process.

Example. Find the final dry-bulb temperature and relative humidity of air washed with recirculated spray water if the air is initially at 95 F dbt with 50 per cent relative humidity as it enters an air washer which has a humidifying efficiency of 85 per cent.

Solution. Figure 7 shows graphically the process. Air at 95 F dbt has 79 F wbt when the relative humidity is 50 per cent. The maximum reduction of the dry-bulb temperature possible is 95 − 79 = 16 F. The actual reduction of dry-bulb temperature will be 16 × 0.85 = 13.6 F so that the final dry-bulb temperature will be 95 − 13.6 = 81.4 F. Following the wet-bulb temperature line, line of constant heat or enthalpy, from 95 F dbt and 50 per cent relative humidity to 81.4 F dbt, indicates that the final relative humidity will be 90 per cent. Air at 81.4 F dbt will tend to be satisfactory from the comfort standpoint, Art. 59, if its relative humidity is below 55 per cent. Consequently the conditions shown by this example are decidedly unsatisfactory from the comfort standpoint. Except in localities where air to be cooled has a low initial relative humidity, evaporative cooling in air-conditioning plants is limited in its applications.

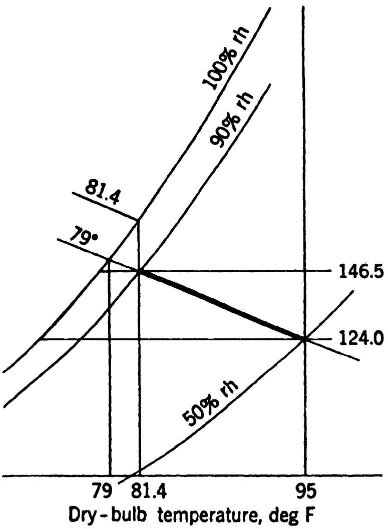


FIG. 7. Evaporative cooling.

46. Drying of Materials.

The removal of moisture from materials which are being processed is an important item in many manufacturing operations. Abstraction of moisture from materials to be dried can be effected by the process of adiabatic saturation using heated air or the hot gases resulting from combustion processes. In a drier contact of the heated air with the moisture to be removed is not so readily made as is the case in apparatus where the water to be evaporated is sprayed into the heated-air stream or where the air is passed over wetted or flooded surfaces. In this operation the total heat or enthalpy of the air and its contained vapor remains constant, and the expression of heat interchange is as given by equation 30.

The weight of dry air required per hour depends upon the weight of moisture to be removed per hour and the moisture pickup per pound of dry air. The hourly weight of air W_a lb equals $W_m \div (w_2 - w_1)$.

The definitions of the symbols are W_m moisture to be removed, lb per hr; w_1 the initial moisture, lb per lb of dry air; and w_2 the final moisture, lb per lb of dry air. In order to arrive at a value for w_2, the relative humidity of the air leaving the drier must be assumed. With the heat and the final relative humidity of the air known the psychrometric chart gives the final dry-bulb temperature of the air.

Example. A drier, operated to remove 8000 lb of moisture per hour from materials within it, has air leaving it with a relative humidity of 80 per cent. The outside air is initially at 60 F dbt and has a relative humidity of 49.4 per cent. The air is heated to a temperature of 175 F by steam coils, and between the heater and the drier air inlet a drop of 5 F occurs in the air temperature. Find the weight of air required and the necessary amount of steam at 5 psig and 0.98 quality to remove the moisture from the materials, both in pounds per hour, when the barometric pressure is 29.92 in. of mercury.

Solution. From the data of Table 6, Chap. 1, the vapor pressure of the outside air may be computed as 0.2563 × 0.494 = 0.1266 psia. This pressure corresponds to a dew-point temperature of 41 F and a humidity ratio of 0.00541 lb of moisture per lb of dry air. The total enthalpy or heat of the mixture at 60 F is

$$h_1 = (0.2401 \times 60) + (0.00541 \times 1088) = 20.29 \text{ Btu per lb of dry air}$$

As the air is first heated to a temperature of 175 F the total enthalpy or heat of the mixture leaving the heater is

$$h_c = (0.2401 \times 175) + (0.00541 \times 1136.2) = 48.16 \text{ Btu per lb of dry air}$$

Likewise the enthalpy of the air-vapor mixture entering the drier is

$$h_2 = (0.2401 \times 170) + (0.00541 \times 1134.2) = 46.95 \text{ Btu per lb of dry air}$$

From the psychrometric chart, Fig. 5 air having a total enthalpy of 46.95 Btu per lb of dry air and a relative humidity of 80 per cent has a dry-bulb temperature of 88.5 F and a humidity ratio of 164 grains or 0.02343 lb of moisture per lb of dry air. The moisture pickup per pound of dry air required is 0.02343 − 0.00541 = 0.01802 lb. The weight of dry air required is 8000 ÷ 0.01802 = 443,950 lb per hr. The heat added to the air-vapor mixture in the coils is 443,950(48.16 − 20.29) = 12,372,885 Btu per hr. If only the latent heat or the enthalpy of evaporation of steam at 19.7 psia is available per pound of steam the hourly weight of steam required is 12,372,885 ÷ (0.98 × 960.6) = 13,143 lb.

47. Air Humidification. Whenever water vapor is added to air, heat energy from some source is required in the process to effect a change of state of the necessary water from a liquid to a vaporous condition. With spray equipment, Fig. 273, the usual schemes employed are: (1) to preheat the air prior to its contact with the water, (2) to heat only the spray water, and (3) to moderately preheat the air and then bring it in contact with heated spray water. Method 2 is never used where the initial temperature of the air may be below 32 F as there is danger of ice formation in the humidifier. Some usage is made of an air humidification process whereby the air is preheated and water vapor produced by the heating of water by steam coils in open pans is added to it.

In the first method of humidification mentioned in connection with water sprays the water is simply recirculated, and heat is not added to it by a separate heater. Preheating the air, which carries an insufficient amount of moisture, increases both its dry- and wet-bulb temperatures. The air passing through water sprays or brought in contact with wetted surfaces tends to become adiabatically saturated as some of the spray water evaporates. The completeness of the process is dependent upon the thoroughness with which the air is brought in contact with the water. Factors involved are the number of banks of spray nozzles used, the effectiveness of the nozzles in breaking up the water into a fine mist, the ratio of air to water handled, the velocity of the air through the spray chamber, the distance the air travels through the sprays, and the insulation of the equipment. Because of inability to satisfy all of the necessary conditions in some commercial apparatus (air washers) the air does not become saturated and a term by-pass factor, Art. 44, is applied in connection with actual performance. Under favorable conditions the air can be made to leave a washer practically saturated and with a wet-bulb temperature very near to that of the entering air. The spray water also assumes a temperature practically the same as the wet-bulb temperature of the entering air. Therefore, the amount of moisture added to air under the conditions of method 1 is dependent upon its wet-bulb temperature. In order to secure the final necessary dry-bulb temperature and relative humidity the air is passed through reheater coils after it leaves the spray chamber. For method 3 the conditions are different, and the process does not involve the principles of adiabatic saturation.

48. Heat Ratios of Air-Vapor Mixtures. The ratio of sensible to total heat either added or abstracted in both air humidification and dehumidification processes is important. The total heat h_t either added or abstracted as the case may be is the sum of the sensible heat h_{sn} and the latent heat h_{ln}. Therefore the ratio of the sensible heat to total heat is

$$sr = \frac{h_{sn}}{h_t} \tag{33}$$

In adiabatic saturation the latent heat involved is the weight of moisture added times the enthalpy of evaporation at the temperature of the spray water or the wet-bulb temperature of the air-vapor mixture leaving the spray chamber. Sensible-heat to total-heat ratios may range from approximately 0.6 to 1.0.

49. Numerical Examples of Humidification by Preheating Air. The various stages of preheating, humidification by adiabatic saturation, and reheating of air are shown pictorially by a skeleton psychrometric chart, Fig. 8.

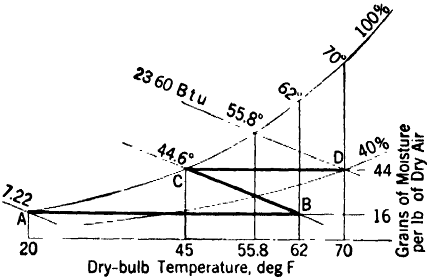


FIG. 8. Air humidification.

Example. Air at 20 F dbt and 100 per cent relative humidity when the barometric pressure is 29.92 in. of mercury (14.696 psia) is to be brought to the condition of 70 F dbt and 40 per cent relative humidity by a process including preheating, adiabatic saturation with recirculated spray water held at the initial wet-bulb temperature of the air, and reheating after it leaves the water-spray chamber. When the humidifying efficiency of the apparatus is 100 per cent find the moisture and the heat added per pound of dry air handled and also the ratio of the sensible to the total heat of the final mixture.

Solution. First locate on the psychrometric chart, Fig. 8, point D representing 70 F dbt and 40 per cent relative humidity. Pass to the left to the saturation curve, and determine C the final dew-point temperature of 44.6 F, which is the dry- and wet-bulb temperature of the saturated air as it leaves the sprays. Follow the wet-bulb temperature line CB representing 44.6 F until it intersects at B with the horizontal line AB drawn through the intersection of the lines denoting 20 F dbt and 100 per cent relative humidity. The processes involved are preheating from 20 to 62 F, along A to B; humidification from B to C, with decrease of the dry-bulb temperature from 62 to 44.6 F; and reheating from C to D to give the final temperature of 70 F dbt with 40 per cent relative humidity. The chart shows 44 grains of moisture and 23.60 Btu per lb of dry air at the final conditions and 16 grains of moisture and 7.22 Btu per lb of dry air initially. The moisture added per pound of dry air is $44 - 16 = 28$ grains or 0.004 lb, and the heat or enthalpy change is $23.60 - 7.22 = 16.38$ Btu per lb. The latent heat or enthalpy of evaporation at 44.6 F is $1068.6 \times 0.004 = 4.27$ Btu per lb. The ratio of sensible to total heat added is $(16.38 - 4.27)/16.38 = 0.74$.

Calculations, employing information given in prior sections of this chapter, can be used to check the foregoing results. Table 9 indicates saturated water-vapor pressures to be 0.0539 psia at 20 F and 0.3628 psia at 70 F. The final vapor pressure at 70 F, based on a relative humidity of 40 per cent, is $0.3628 \times 0.4 = 0.1451$ psia which is the saturation pressure corresponding to a dew-point temperature of 44.57 F. Final and initial humidity ratios w_2 and w_1 are $w_2 = (0.622 \times 0.1451)/(14.696 - 0.1451) = 0.0062$ lb or 43.4 grains per lb, and $w_1 = (0.622 \times 0.0539)/(14.696 - 0.0539) = 0.00229$ lb or 16.03 grains. The moisture added is $43.4 - 16.03 = 27.37$ grains per pound of dry air. The enthalpies per pound of dry air are: $h_1 = (0.2401 \times 20) + (0.00229 \times 1069.8) = 7.25$ where h_g at 20 F equals

1060.8 + (0.45 × 20) = 1069.8 Btu per lb; $h_2 = (0.2401 \times 70) + (0.0062 \times 1092.3) = 23.58$ Btu per lb of dry air and contained water vapor. The heat added or the enthalpy change per pound of dry air is 23.58 − 7.25 = 16.33 Btu.

When the design and operation of the humidifier are such that complete adiabatic saturation of the air is not possible the air must be preheated to a higher temperature, if the required final conditions are to be obtained.

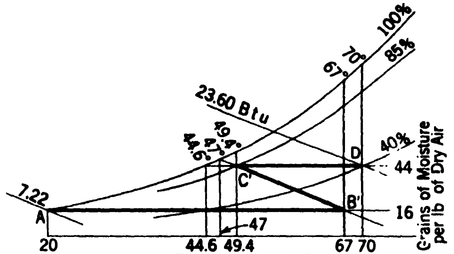


FIG. 9. Air humidification.

Example. The initial and final conditions of air are to be the same as in the prior problem, with the exception that the relative humidity of the air entering the reheater is the highest obtainable with the equipment used. Find the temperature to which the air must be preheated and also the humidifying efficiency of a washer was used.

Solution. The intersection of the constant weight of vapor line for 70 F dbt and 40 per cent relative humidity and the curve for 85 per cent relative humidity of Fig. 9 fixes the temperatures of the air leaving the humidifying chamber as 49.4 F dbt and 47 F wbt at point *C'*. Moving along the wet-bulb temperature line of 47 F to the constant weight of vapor line at *B'* it will be noted that the preheated air must have 67 F dbt. Hence the path of the process in preheating from 20 F dbt and 100 per cent relative humidity is along *AB'* to 67 dbt; humidification along *B'C'* to give a condition of 85 per cent relative humidity; and reheating along *C'D* to give the desired final state. The theoretical reduction of the dry-bulb temperature as water vapor is added is 67 − 47 = 20 F. The actual reduction of the dry-bulb temperature is 67 − 49.4 = 17.6 F. Hence the humidifying efficiency is (17.6 × 100) ÷ 20 = 88 per cent. The final dew-point temperature and the amounts of moisture and heat added are as in the previous example.

50. Humidification of Preheated Air by Use of Heated Water.

When spray water is heated by an independent heater it is maintained at a temperature corresponding to the dew-point of the air at the final desired conditions.

Example. Air at 20 F dbt and 100 per cent relative humidity after being preheated to 40 F dbt is to be humidified and heated to give the final conditions of 70 F dbt and 40 per cent relative humidity when the barometric pressure is 29.92 in. of mercury. Find the heat supplied by the air and water heaters and the moisture added per pound of dry air, if the make-up water enters the water heater at a temperature of 40 F.

Solution. A study of the skeleton psychrometric chart, Fig. 10, indicates that the initial and final conditions are the same as those of the examples of Art. 49 as regards dew-point temperature, humidity ratio, moisture added, and change of enthalpy between points *A* and *D*. The process includes preheating from *A* to

B, addition of moisture along some not-too-well-defined path between *B* and *C*, and reheating from *C* to *D*. Sometimes the path between the points *B* and *C* is shown as the dotted straight line joining the two. By the chart under initial conditions each pound of dry air has a humidity ratio w_1 of 16 grains and an enthalpy h_1 of 7.22 Btu per lb. For the final conditions, w_2 is 44 grains, and h_2 equals 23.60 Btu per lb of dry air. Therefore in the completed process the moisture increase per pound of dry air is 44 − 16 = 28 grains or 0.004 lb, and the change of enthalpy is 23.60 − 7.22 = 16.38 Btu per lb. Make-up water entering the system has an enthalpy of the liquid which amounts to 8.05 Btu per lb at 40 F. Since the value of 16.38 Btu per lb includes heat from all sources the net heat supplied by the heaters is 16.38 − (0.004 × 8.05) = 16.35 Btu per lb of dry air and its contained vapor. The ratio of the sensible heat to the total heat added is 0.74, as in the first example of Art. 49.

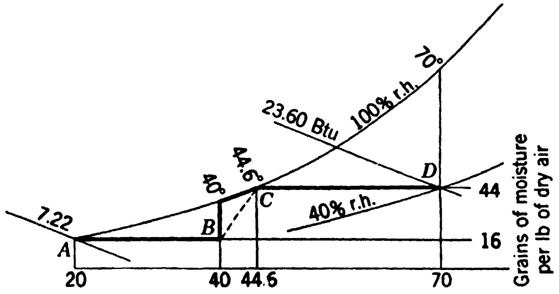
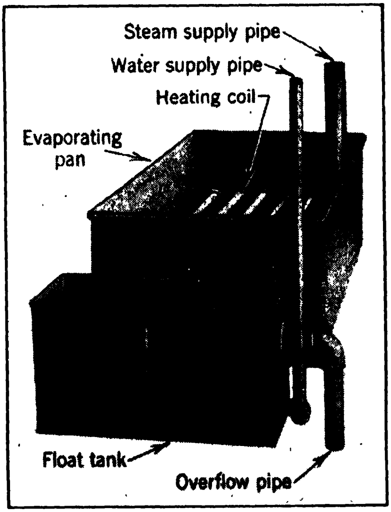


FIG. 10. Air humidification with heated spray water.



(Johnson Service Co.)

FIG. 11. Pan humidifier.

Thus far the discussion of air humidification has involved some form of spray equipment similar to the air washer of Figs. 272 and 273. A simple steam-coil-heated open-pan humidifier, as illustrated by Fig. 11, permits the vaporization of water at 212 F at the standard atmospheric pressure of 14.696 psia. With such equipment the operations are as indicated by the skeleton chart of Fig. 12 for the data of the following example.

Example. Saturated air at 20 F is to be preheated, humidified by means of open-pan equipment, and then reheated to give a relative humidity of 40 per cent and a dry-bulb temperature of 70 F. Determine the necessary procedure if the water vapor is supplied at 212 F.

Solution. The prior solution of this article indicated that under the initial conditions the dry air has 16 grains (0.00228 lb) of water vapor and 7.22 Btu per lb of dry air. The final conditions of 40 per cent relative humidity and 70 F dbt involve a dew point temperature of 44.6 F, 44 grains (0.00628 lb) of water vapor, and 23.6 Btu per lb of dry air. The enthalpy of dry saturated water vapor at 212 F is 1150.4 Btu per lb; therefore the heat added per pound of dry air as the vapor mixes with it is $1150.4(0.00628 - 0.00228) = 4.6$ Btu. Dependent upon the initial dry-bulb temperature of the mixture entering the humidifier the process may start at either B, B', B'', or at any intermediate condition between B and B'' of Fig. 12. The dew-point temperature for the final desired state is 44.6 F at C where the enthalpy is $h_c = (0.24 \times 44.6) + 0.00628[1060.8 + (0.45 \times 44.6)] = 17.5$ Btu per lb. When B represents the initial conditions under which the air to be moistened enters the humidifier its existent enthalpy is $17.5 - 4.6 = 12.9$ Btu per lb. In order to satisfy this condition the air must be preheated from 20 F along line AB to B. Here the required minimum temperature t_b may be calculated from $12.9 = 0.24t_b + 0.00228\,(1060.8 + 0.45t_b)$. The minimum temperature for starting the humidification process along line BC is 43.5 F. After the air has received all of the required vapor between B and C additional heat amounting to $0.24(70 - 44.6) + 0.00628 \times 0.45(70 - 44.6) = 6.16$ Btu must be added in a separate heater along CD. The calculated total enthalpy at D is $17.5 + 6.16 = 23.66$ which checks the 23.6 Btu per lb given by Fig. 12.

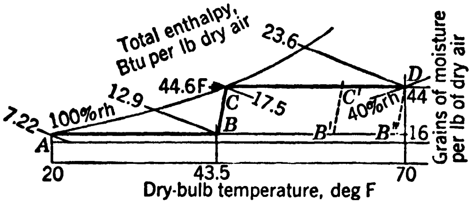

Total enthalpy, Btu per lb dry air
23.6
12.9
44.6F
17.5
7.22
100%rh
C
B
A
C′
40%rh
B′
B″
D
44
16
20
43.5
70
Dry-bulb temperature, deg F
Grains of moisture per lb of dry air

FIG. 12. Air humidification with pan-type unit.

When for any reason the temperature of the initial mixture is 60 F at B' its enthalpy is $(0.24 \times 60) + 0.00228[1060.8 + (0.45 \times 60)] = 16.82$ Btu per lb. The enthalpy at C' is $16.82 + 4.60 = 21.42$ Btu per lb. The dry-bulb temperature $t_{c'}$ at C' is calculated from $21.42 = 0.24t_{c'} + 0.00628(1060.8 + 0.45t_{c'})$ as 61 F. Additional heat to be added from C' to D equals $0.24(70 - 61) + 0.00628 \times 0.45(70 - 61) = 2.19$ Btu and the enthalpy at D is $21.42 + 2.19 = 23.61$ Btu per lb by this method.

The maximum temperature $t_{b''}$, for these particular conditions, at which the air may enter the humidifier is computed from $23.60 - 4.60 = 19 = 0.24t_{b''} + 0.00228(1060.8 + 0.45t_{b''})$. This temperature is 68.8 F which is 1.2 F less than 70 F the required terminal condition.

These calculations indicate that the process within the humidifier follows a path fixed by the initial dry-bulb temperature of the air-vapor mixture entering it. Calculated increases of dry-bulb temperature within the humidifier are 1.1, 1.0 and 1.2 F respectively. The assumption is made that only heat used to vaporize water at 212 F is used in the humidification process and that there are no other sources of heat within the humidifier. This method of calculation may be applied when the vapor temperature is less than 212 F.

51. Air Dehumidification. The processes of moisture removal from air containing water vapor include either cooling the mixture to a temperature whereby the required amount of vapor is condensed or by the use of sorbents. Sorbents are further classified as absorbents and adsorbents. Calcium chloride and lithium chloride are absorbents, and when water vapor unites with them because of vapor pressure differences the dry-bulb temperature of the air rises as water is absorbed from it with a transformation of latent to sensible heat. Adsorbents include silica gel and activated alumina. There is no chemical reaction with adsorbents, but because of a difference of vapor pressure within and without them they pick up water vapor until they become inactive. Such materials are reactivated by driving moisture from them by the application of heat. When sorbents are used to remove moisture from air the dry-bulb temperature of the air rises as a transformation of latent heat to sensible heat occurs. Often this is of advantage in air cooling as sensible heat at a higher temperature level above that of the cooling agent is easier to remove because of more favorable conditions of heat transfer than exist when the heat is in the latent form, and the air-vapor mixture has a temperature which is closer to that of the coolant. In all cases where sorbents are used for the purpose of dehumidification the ratio of the sensible heat to the total heat is increased, as water vapor is removed by the action of the sorbent.

52. Dehumidification of Air by Cooling Processes. Water vapor may be removed from air by passing it either through chilled-water sprays or over cooling coils. Where spray water is used it must be maintained at a temperature below the dew-point temperature of the air coming in contact with it. Either ice or mechanical refrigeration is necessary to cool the spray water when the water is to be used again. Occasionally cold well water is available for air cooling and dehumidification; in such cases when it is used either in sprays or cooling coils the warmed water is allowed to waste to the sewer which is quite undesirable where water supplies are limited. Where air cooling and dehumidification are accomplished by means of coil surfaces the cooling medium held within the coil or coils must be maintained at a temperature below the dew-point temperature of the cooled and dehumidified air.

The process of adiabatic saturation cannot be reversed in the dehumidification of air as heat removal from the air-cooling medium takes place outside of the spray chamber or the cooling coils.

The heat removed per pound of dry air in cooling a mixture of air and water vapor below its dew-point temperature includes the sensible heat of the dry air $c_{pa}(t_1 - t_2)$, the heat of superheat of the vapor $c_{ps}w_1(t_1 - t_s)$, the heat removed from the saturated vapor $w_2(h_{gs} - h_{g2})$, the heat of evaporation at the average temperature between the initial dew point and the final air temperature $(w_1 - w_2)h_{fga}$, and the enthalpy of the liquid between the average temperature at which moisture condensation occurs and the final air temperature $(w_1 - w_2)(h_{fa} - h_{f2})$. The definitions of the symbols are respectively: t_1 and t_2 the initial and final dry-bulb temperatures, w_1 and w_2 the initial and final weights of moisture per pound of dry air, t_s the dew-point temperature of the initial air, h_{fga} the heat of evaporation at the average temperature at which moisture is removed, h_{fa} the enthalpy of the liquid at the average temperature of moisture removal, and h_{f2} the enthalpy of the liquid at the final air temperature.

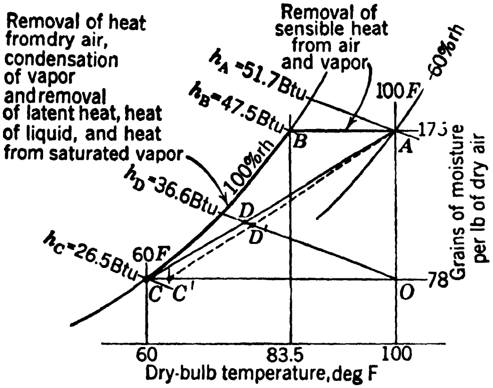

Removal of heat fromdry air, condensation of vapor andremoval of latent heat, heat of liquid, and heat from saturated vapor
Removal of sensible heat from air and vapor
h_A=51.7Btu
h_B=47.5Btu
h_D=36.6Btu
h_C=26.5Btu
60%rh
100%rh
100F
60F
A
B
D
D'
C
C'
O
175
78
Grains of moisture per lb of dry air
60
83.5
100
Dry-bulb temperature, deg F

FIG. 13. Air dehumidification with a spray cooler.

53. Air Cooling and Dehumidification with Water Sprays. The theoretical performance of a spray humidifier is shown by Fig. 13. Pieces of spray equipment, which are properly designed and carefully operated, give results which are close to those shown by Fig. 13. In Fig. 13 the removal of sensible heat is along the constant weight of vapor line *AB* until the dew-point temperature is reached at *B*, where condensation of water vapor begins. As condensation occurs along the saturation curve from *B* to *C*, both sensible and latent heat are removed. The total heat removed is the sum of the sensible and the latent heats. As condensation takes place the ratios of the sensible to the total heat change. Enthalpies of the mixture for conditions *A*, *B*, *C*, and *D* are h_A, h_B, h_C, and h_D. The sensible heat removed is $h_D - h_C$, and the latent heat abstracted is $h_A - h_D$. The sensible-heat ratio or sensible-heat factor (*SHF*) is $(h_D - h_C) \div (h_A - h_C)$, which is an important item in air-conditioning problems. The line *CA* is a load-ratio line, and parts *CD* and *DA* represent proportionally the amounts of sensible and latent heat removed.

Example. Air at 100 F dbt and 60 per cent relative humidity is to be cooled and dehumidified by passing it through a cold-water spray when the barometric pressure is 29.92 in. of mercury. The spray water, at a suitable temperature, is adequate in quantity for air-washer operation, and the velocity of air flow is not excessive. Find the final relative humidity of the air, the moisture removed, the heat removed, and the ratio of sensible to total heat removed when the final temperature is 60 F.

Solution. The psychrometric chart, Fig. 13, indicates w_A as 175 grains of moisture and h_A as 51.7 Btu per lb of dry air and its contained moisture at the initial state. Sensible heat removal along line *AB* with a constant weight of vapor w_A equal to 175 grains occurs until the initial dew-point temperature, as indicated by the graph, is produced at 83.5 F. The air remains saturated with 100 per cent relative humidity as it is cooled, with the removal of both sensible and latent heat, to a final temperature of 60 F. Under the final conditions, with air at 100 per cent relative humidity, w_C is 78 grains and h_C is 26.5 Btu per lb of dry air. The moisture removed is $w_A - w_C = (175 - 78) = 97$ grains or 0.0139 lb per lb of dry air, and the total abstraction of heat h_T is $h_A - h_C$ or $(51.7 - 26.5) = 25.2$ Btu per lb. The sensible heat removed is $h_D - h_C$ or $(36.6 - 26.5) = 10.1$ Btu per lb. The ratio of the sensible heat to the total heat *SHF* is $10.1 \div 25.2 = 0.4$. When this ratio is feasible the operation of the unit may be considered as satisfactory.

The design and the operating conditions of a spray dehumidifier may not give the performance required to produce the state *C* in Fig. 13. When the terminal condition is at C' the humidity ratio will be the correct one but the air will not be saturated and its dry-bulb temperature will be higher than at *C*. The path of the process between *A* and C' will not be as previously assumed. Irrespective of the path followed the total heat removed will be $h_A - h_{C'}$. The load-ratio line will be $C'A$. With a higher dry-bulb temperature at C' than at *C* the *SHF* will be different and a greater amount of air will be required to produce the desired conditions within a space.

54. Air Cooling and Dehumidification with Coils. When coil surfaces are used a number of factors such as surface temperatures; areas; depth of sections; velocities of air flow; the refrigerant, its temperature, and direction of flow; the condition of the cooling surfaces, that is, wet or dry; and the by-passing of air over the coil surfaces affect their performance.[5, 6]

The following discussion, dealing with finned cooling coils, involves consideration of the humidity method of determining their performances. Figure 14 is a skeleton representation of the use of either a psychrometric chart as described in this book or a similar non-logarithmic one. When the inlet, and the required exit-air conditions are as indicated by *A* and *D,* Fig. 14, the load-ratio line *AD* makes an angle with the horizontal. The slope of the line is dependent upon the existing conditions. Investigations have indicated that the path

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[5] "Performance of Surface Coil Dehumidifiers for Comfort Air Conditions," by G. L. Tuve and J. Seigel, *ASHVE Trans.*, Vol 44, 1938.

[6] "Air Cooling Coil Problems and Their Solutions," by L. G. Seigel, *ASHVE Trans.*, Vol. 51, 1945.

traveled by an air-vapor-mixture process, for any particular condition, may be along the dotted pursuit curve *A*1 without removal of sensible heat entirely along *AB* and the removal of both sensible and latent heat with constant relative humidity from *B* to 1. In any event the heat and moisture removals between *A* and *D* are the same whatever course the process follows. The same is true for another path between *A* and *C*.

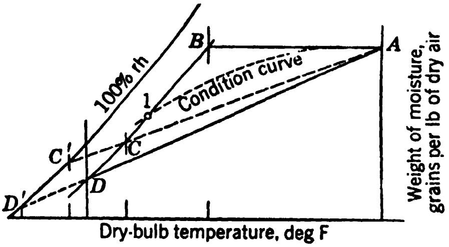
B
A
100% rh
Condition curve
1
C'
C
D'
D
Dry-bulb temperature, deg F
Weight of moisture, grains per lb of dry air

FIG. 14. Air dehumidification with a cooling coil.

All points located on a relative humidity curve of a psychrometric chart are uniformly horizontally distant from the saturation curve as shown by *BD* of Fig. 14. The path *ABCD* does not actually indicate the condition of the air-vapor mixture as it passes through a cooling coil; it does represent on a chart the possible final condition of air leaving a coil operating with decreasing refrigerant (or surface) temperatures.

Lines *AC* and *AD* when extended to the 100 per cent relative humidity curve locate *C'* and *D'* close to the apparatus dew-point temperatures. Apparatus dew-point temperatures are those of the surface areas of the spray-water droplets within spray-type humidifiers and the cooling-surface temperatures of coil-type dehumidifiers. Apparatus dew-point temperatures and the design of cooling coils are discussed in Arts. 409, 410, and 421. When lines *AC* and *AD* are extended and do not intersect the saturation curve the design conditions must be somewhat different than when the apparatus dew point can be located graphically. The discussions of Arts. 409, 410, and 421 indicate how to find the required terminal conditions at either *C* or *D*. The following example illustrates the amounts of heat and moisture to be removed when the exit-air-vapor-mixture conditions have been fixed for a dehumidifier.

Example. Air at 95 F dbt and 78 F wbt is to be cooled and dehumidified by passing it over a refrigerant-filled coil which has sufficient depth with the temperature of the refrigerant used to give a final condition of 60 F dbt and a relative humidity of 90 per cent. Find the heat and the amount of moisture removed per pound of dry air. What is the apparatus dew-point temperature?

Solution. The skeleton psychrometric chart, Fig. 15, indicates that the initial moisture is 118 grains per lb of dry air with a total enthalpy of 41.5 Btu per lb, a relative humidity of 47.8 per cent, and a dew-point temperature of 72 F. Sensible heat is shown to be removed at a constant humidity ratio of 118 grains until

90 per cent relative humidity is reached at 75 F dbt. Following the 90 per cent relative humidity line until 60 F dbt is reached the final humidity ratio is 69 grains with a total enthalpy of 25.1 Btu per lb of dry air and contained moisture. The moisture removal is 118 − 69 = 49 grains, and the heat removal is 41.5 − 25.1 = 16.4 Btu per lb of dry air. The heat-load ratio of sensible heat to total heat removed is (33.8 − 25.1) ÷ (41.5 − 25.1) = 0.53. The apparatus dew-point temperature is 52.5 F.

As in the case of spray equipment there is a by-pass action involved. Unless the cooling section has a depth of several tubes the air contact with cooling surfaces cannot be as thorough with coils as is the case where spray equipment breaks the cooling medium into fine droplets. Each additional row of tubes in a coil adds frictional resistance to air flow which the fan must overcome with increased power requirements. Generally a satisfactory amount of cooling and dehumidification can be accomplished with a coil, operating with reasonable air-friction losses, to give final air relative humidities of 90 to 95 per cent.

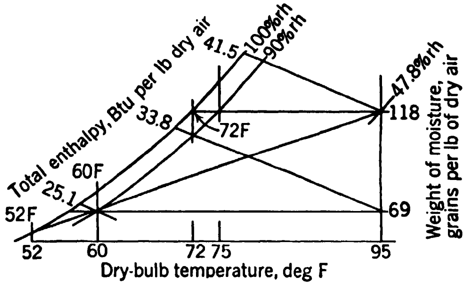

FIG. 15. Dehumidification of an air-vapor mixture with a cooling coil.

55. Air Dehumidification or Dehydration by Use of Sorbents. *Adsorbents* condense water vapor on the internal surfaces of their porous materials which are in solid forms and include such substances as silica gel, activated alumina, and activated carbon. Desirable adsorbents must have a suitable vapor-pressure characteristic, reasonable first cost, sufficient moisture-adsorbing ability to avoid excessive bed dimensions, chemical stability, resistance to breakdowns from handling and reactivation, and be capable of being reactivated for further use, after they have been rendered ineffective by moisture adsorption, by the application of heat at reasonable temperatures. Dehumidification of air-vapor mixtures is accomplished with adsorbents by either drawing or blowing the mixture through a bed of the material where the water vapor is caught and retained in the pores of the agent used. The action is due to surface condensation in the pores and the difference of the vapor pressure of water condensed in the pores and the vapor pressure of the air-vapor mixture. The process continues until the state of equilibrium exists between the vapor pressures. There are no chemical changes in the process of adsorption, but heat is liberated to warm the air handled and the adsorbent bed. The heat of wetting for silica gel is about 200 Btu per lb of the material, and the total heat adsorption of this material is approximately 1290 Btu per lb of water removed.

The most commonly used *absorbent* in solid form is calcium chloride which has small and limited applications. Liquid absorbents are solutions of chlorides and bromides, such as solutions of calcium chloride, lithium bromide, and lithium chloride. Ethylene glycol is also another absorber of water vapor. Absorbents change both physically and chemically during the process of taking up water vapor. Where absorbents are used in liquid form. good contact can be obtained by passing the air-vapor mixture to be dehydrated over large surface areas of the liquid which are obtained by breaking the absorbent up into a fine mist by the means of nozzles. In the absorption process the water vapor gives up its latent heat of vaporization or condensation, and there is some heat liberated by the mixing process. In any event the dry-bulb temperature of the air-vapor mixture which is dehumidified is increased.

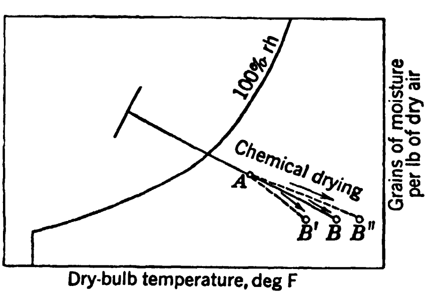
100% rh
Chemical drying
A
B' B B''
Dry-bulb temperature, deg F
Grains of moisture per lb of dry air

FIG. 16. Dehumidification with a sorbent.

For either the process of adsorption or absorption of water vapor from air the action is not the reverse of adiabatic saturation of air at a practically constant wet-bulb temperature. Where the drying agent is soluble in water the process takes place above the wet-bulb line AB, Fig. 16, along AB'', or below it along AB', depending upon whether heat is liberated or absorbed when the agent dissolves in water. When an appreciable amount of heat is retained by the adsorber the process takes place along the line AB'.

56. The Mixing of Quantities of Air or Air-Vapor Mixtures. Quite often in air-conditioning calculations it is desirable to be able to find the resultant temperature, the humidity ratio, and the enthalpy of the resulting mixture when two or more quantities of air are combined. Thus, using an equation based on weights and temperatures, the resulting temperature of a mixture of two quantities of dry air may be had from

$$c_{pa}t_r(W_{a1} + W_{a2}) = c_{pa}(t_1W_{a1} + t_2W_{a2})$$

$$t_r = \frac{t_1W_{a1} + t_2W_{a2}}{W_{a1} + W_{a2}} \tag{34}$$

where t_r = resultant temperature, deg F.
t_1 = temperature of W_{a1} lb of air, deg F.
t_2 = temperature of W_{a2} lb of air, deg F.
W_{a1} = weight of air at temperature t_1, lb.
W_{a2} = weight of air at temperature t_2, lb.
c_{pa} = mean specific heat of air at constant pressure, 0.2401.

When weights of water vapor W_{m1} and W_{m2} in pounds are involved with dry air the following equations may be written

$$c_{pa}t_r(W_{a1} + W_{a2}) + c_{ps}t_r(W_{m1} + W_{m2}) = c_{pa}(t_1W_{a1} + t_2W_{a2}) + c_{ps}(t_1W_{m1} + t_2W_{m2})$$

$$t_r = \frac{c_{pa}(t_1W_{a1} + t_2W_{a2}) + c_{ps}(t_1W_{m1} + t_2W_{m2})}{c_{pa}(W_{a1} + W_{a2}) + c_{ps}(W_{m1} + W_{m2})} \tag{35}$$

in which c_{ps} is the mean specific heat of water vapor 0.45. Except where the amounts of water vapor mixed with the dry air are large, little difference in the value of t_r as calculated by equations 34 and 35 will be obtained.

The resultant enthalpy in Btu per pound of dry air and its water-vapor content may be derived from $h_r(W_1 + W_2) = h_1W_1 + h_2W_2$ as

$$h_r = \frac{h_1W_1 + h_2W_2}{W_1 + W_2} \tag{36}$$

where h_r = the resultant total enthalpy, Btu per lb of dry air and its vapor.
h_1 = the total enthalpy, Btu per lb of W_1 lb of dry air and its vapor.
h_2 = total enthalpy, Btu per lb of W_2 lb of dry air and its vapor.

The weight of water vapor per pound of dry air after mixing two weights, W_{a1} and W_{a2}, each having humidity ratios w_1 and w_2 lb of water vapor per pound of dry air, is derived from $w_r(W_{a1} + W_{a2}) = W_{a1}w_1 + W_{a2}w_2$ whereby the resultant humidity ratio is

$$w_r = \frac{W_{a1}w_1 + W_{a2}w_2}{W_{a1} + W_{a2}} \tag{37}$$

Example. Twenty-thousand pounds of dry air having 100 F dbt, a humidity ratio of 116 grains per lb of dry air, and an enthalpy of 42.32 Btu per lb of dry air

and its contained water vapor are mixed with 50,000 lb of dry air having 70 F dbt, a humidity ratio of 66 grains, and an enthalpy of 27.11 Btu per lb of dry air and its vapor. Find the resultant dry-bulb temperature, the humidity ratio of the new mixture, and its enthalpy, when the barometer reads 29.92 in. of mercury.

Solution.

$$t_r = \frac{0.24[(100 \times 20{,}000) + (70 \times 50{,}000)] + 0.45[(100 \times 20{,}000 \times 0.01657) + (70 \times 50{,}000 \times 0.00943)]}{0.24(20{,}000 + 50{,}000) + 0.45[(20{,}000 \times 0.01657) + (50{,}000 \times 0.00943)]} = 78.6+ \text{ F}$$

$$h_r = \frac{[(42.32 \times 20{,}000) + (27.11 \times 50{,}000)]}{(20{,}000 + 50{,}000)} = 31.45 \text{ Btu}$$

$$w_r = \frac{[(20{,}000 \times 116)] + (50{,}000 \times 66)]}{(20{,}000 + 50{,}000)} = 80.3 \text{ grains}$$

The foregoing case of the mixing of two quantities of air-vapor mixtures may be shown graphically on a psychrometric chart as in Fig. 17. Point B represents the final conditions after the stated quantities of air-vapor as represented by the data at A and C have been mixed.

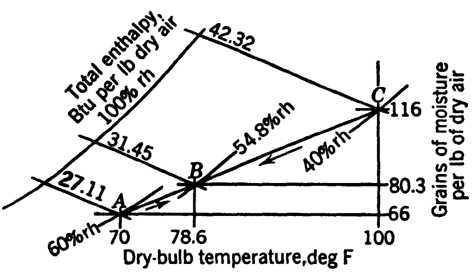

FIG. 17. Mixing of air-vapor quantities.

57. Weight of Air Required as a Carrier of Heat. In both heating and cooling operations air is used as an agent to either give up or receive heat. Generally speaking, the weight of air required is dependent upon the amount of heat to be carried and the amount of sensible heat given up or picked up per pound of dry air. For air giving up heat the amount per pound of dry air is

$$c_{pa}(t_i - t_f) + c_{ps}w_i(t_i - t_f)$$

where c_{pa} = the mean specific heat of air at constant pressure.

t_i = the initial temperature of the air.

t_f = the final temperature of the air.

c_{ps} = the mean specific heat of the superheated water vapor.

w_i = the humidity ratio per pound of dry air.

For many calculations the portion $c_{ps}w_i(t_i - t_f)$ of the foregoing equation may be ignored, and the weight of dry air necessary to give up H heat units, Btu per hour, is

$$W_a = \frac{H}{c_{pa}(t_i - t_f)} \tag{38}$$

Where cooling is involved and the air picks up heat in a space the equation becomes

$$W_a = \frac{H}{c_{pa}(t_f - t_i)} \tag{39}$$

PROBLEMS

1. Find by calculation the volume and the density of 10 lb of dry air when it exists at −58 F dbt and a barometric pressure of 24.5 in. of mercury.

2. Calculate the weight of water vapor required to saturate one pound of dry air when at 78 F dbt and under a barometric pressure of 27.43 in. of mercury. Find the specific volume and the density of the mixture when saturated.

3. Calculate the humidity ratio of saturated air at 78 F when the barometric pressure is 28.5 in. of mercury. Calculate the specific volume and the density of the mixture.

4. Calculate the total enthalpy of saturated air at 72 F and a barometric pressure of 29.5 in. of mercury. Express as Btu per pound of dry air.

5. Compute the enthalpy of saturated air at 60 F and a barometric pressure of 26.5 in. of mercury. Express as Btu per pound of dry air.

6. Air has 88 F dbt and 72 F wbt when the barometric pressure is 28.8 in. of mercury. Find its relative humidity, the percentage of saturation, and the dew-point temperature.

7. Find the relative humidity, the percentage of saturation, and the dew-point temperature of air which has 90 F dbt, 70 F wbt, and is under a barometric pressure of 29 in. of mercury.

8. Calculate the humidity ratio and the enthalpy per pound of dry air for the conditions of problem 6.

9. Determine the humidity ratio and the enthalpy per pound of dry air for the conditions of problem 7.

10. Calculate all of the psychrometric properties of saturated air for the conditions of 92 F and a barometric pressure of 29.5 in. of mercury.

11. By the use of the psychrometric chart, Fig. 5, find all the properties of an air-vapor mixture existing under a barometric pressure of 29.92 in. of mercury, 88 F dbt, and 76 F wbt.

12. Show a comparison of the actual psychrometric properties of an air-vapor mixture existing at a barometric pressure of 27 in. of mercury, 80 F dbt, and 60 F wbt with those read from a standard psychrometric chart for 29.92 in. of mercury when the temperatures are the same in both cases.

13. Air under a barometric pressure 29.92 in. of mercury has 80 F dbt and 60 F wbt. Determine by calculations the final dry-bulb temperature, the increase of the humidity ratio, and the heat transformation per pound of dry air if the air is adiabatically saturated. Check the calculations by use of a standard psychrometric chart for a pressure of 29.92 in. of mercury.

14. Calculate the increase in the humidity ratio and the heat transformation per pound of dry air for the temperatures mentioned in problem 13 if the air is adiabatically saturated when the barometric pressure is 27.3 in. of mercury.

15. A piece of spray apparatus was used to partially adiabatically saturate air, in processing an air-vapor mixture which entered it at 80 F dbt and 65 F wbt, to give a final 68 F dbt condition. Find the humidifying efficiency and the by-pass factor of the apparatus.

16. Find the humidifying efficiency and the by-pass factor for spray equipment, using unheated recirculated spray water, which gave a final condition of 77 F dbt with air which entered the apparatus at 88 F dbt and 76 F wbt.

17. Ten-thousand pounds of moisture are to be removed each hour from materials which are to be dried by passing heated air over them. Air from outdoors at 70 F dbt and a relative humidity of 40 per cent is heated to 175 F and passed to the drier at 145 F dbt. The final relative humidity of the partially saturated air leaving the drier is 90 per cent. Steam at 20 psia and 0.97 quality is used in the air heater with the condensate leaving it at steam temperature. Find the weights of wet steam and dry air required per hour if the barometric pressure is 29.92 in. of mercury.

18. Air is taken from outdoors at 50 F dbt and 60 per cent relative humidity and heated by steam at 22 psia and 0.98 quality to a temperature of 165 F dbt. The air then enters the drier where it picks up 50,000 lb per hr of water vapor to give it a relative humidity of 88 per cent. Find the weights of air and steam required in pounds per hour when the barometric pressure is 29 in. of mercury.

19. Air is to be humidified by complete adiabatic saturation. The air is initially at 25 F dbt with a dew-point temperature of 20 F. The air is to be preheated to the proper temperature, humidified, and reheated to give a final condition of 68 F dbt and 50 F wbt. Find the amount of water vapor and the heat added during the process. The barometric pressure is 29.92 in. of mercury.

20. Air is to be brought to a final condition of 71 F dbt and 35 per cent relative humidity from an initial condition of 25 F dbt and 90 per cent relative humidity by the processes of pre-heating, partial adiabatic saturation to give a relative humidity of 95 per cent, and reheating to the terminal condition when the barometric pressure is 29.92 in. of mercury. Find the moisture and the heat added during the operations.

21. Air at an initial condition of 22 F dbt and 90 per cent relative humidity is to be brought to 72 F dbt and 35 per cent relative humidity by being preheated to 35 F dbt and then humidified by being washed with heated spray water held at a temperature corresponding to that of the dew point at the final conditions. Reheating is to be used to give the final desired temperature and relative humidity. Find the heat and moisture added per pound of dry air and the ratio of the sensible to the total heat added. The barometric pressure is 28.95 in. of mercury.

22. Air at 83 F dbt and 63 F wbt is to be cooled to produce saturated air at 50 F when the barometric pressure is 29.92 in. of mercury. The spray water is at a temperature below the final dew-point temperature. Calculate the amount of moisture and the heat removed per pound of dry air.

23. A spray-type air cooler reduces the dry-bulb temperature of an air-vapor mixture from 95 to 75 F when the initial relative humidity is 60 per cent and the final relative humidity is 95 per cent. The barometric pressure is 28.75 in. of mercury. Find the heat and moisture removed per pound of dry air.

24. A cooling coil is supplied with an air-vapor mixture at 90 F dbt and 40 per cent saturation. The apparatus discharges the air with a saturation of 90 per cent and 55 F dbt. Find the amount of moisture and heat removed per pound of dry air and the sensible-heat factor when the barometric pressure is 29.92 in. of mercury.

25. A cooling coil operates to give at its outlet a 65 F dbt together with a relative humidity of 85 per cent when the air-vapor mixture under initial conditions has 95 F dbt and a relative humidity of 50 per cent; the barometric pressure is 29.92 in. of mercury. Calculate the moisture and the heat removed per pound of dry air and the *SHF*.

26. Eighty thousand pounds of dry air at 80 F dbt and 60 F wbt, are mixed with 25,000 lb of dry air at 50 F dbt and 40 F wbt. Calculate the final air temperature, the humidity ratio, and the final enthalpy in Btu per pound after mixing has taken place if the barometric pressure is 29.92 in.of mercury.

27. Forty thousand pounds of dry air with 74 F dbt and 58 F wbt are mixed with 50,000 lb of dry air at 85 F dbt and 75 F wbt. Find the resulting dry-bulb temperature, the humidity ratio, and the enthalpy per pound of dry air when the barometer is 28.75 in. of mercury.

28. A room has heat losses amounting to 150,000 Btu per hr when its air temperature is maintained at 70 F. Air is introduced into the room at 175 F dbt when the barometric pressure is 29.92 in. of mercury. Find the weight of dry air required per hour to serve as a carrier of heat and the volume of the air supplied per minute measured at 175 F.

29. The heat losses from a factory building amount to 2,000,000 Btu per hr when the room temperature is held at 65 F dbt. Air for heating and ventilating purposes is supplied to the building at 135 F when the barometric pressure is 28.57 in. of mercury. Find the weight of air required per hour and the volumes of the air necessary per minute when measured at 65 F and 135 F.

CHAPTER 3

FACTORS AFFECTING HUMAN COMFORT

58. Foreword. If the desired results are to be obtained with either winter or summer air-conditioning systems, the designer of such plants must be cognizant of a number of factors which physiologically affect human comfort when such is involved. The factors include effective temperature, the production and regulation of heat in the human body, heat and moisture losses from the human body, air motion, the effects of cold and hot surfaces within the spaces considered, and the stratification of air.

59. Effective Temperature. Comfort conditions for individuals, excluding air odors and cleanliness, are dependent upon the dry- and wet-bulb temperatures of the air and its rate of motion. These three items collectively in various combinations produce conditions of either comfort or discomfort. Discomfort may come from unsatisfactory sensations of either heat or cold.

The arbitrary index which combines in a single value the degree of warmth or cold felt by the human body in response to the air temperature, moisture content, and motion is termed *effective temperature*. Effective temperature cannot be measured directly but is fixed as the temperature of saturated still air (velocity 15–25 fpm due to natural turbulence) which induces the same sensations of warmth or coolness as those produced by the air surrounding a person.

Figure 18 illustrates the ASHVE Comfort Chart[1] as developed by Houghten, Yaglou, and Drinker. This chart is for still-air conditions, and the diagonal temperature lines represent effective temperatures. Examination of the graph reveals that a number of combinations of wet- and dry-bulb temperatures and relative humidity will produce the same effective temperature. Two zones of comfort are shown, i.e., winter and summer comfort conditions which overlap to some extent. The optimum effective temperature for winter weather is shown to be

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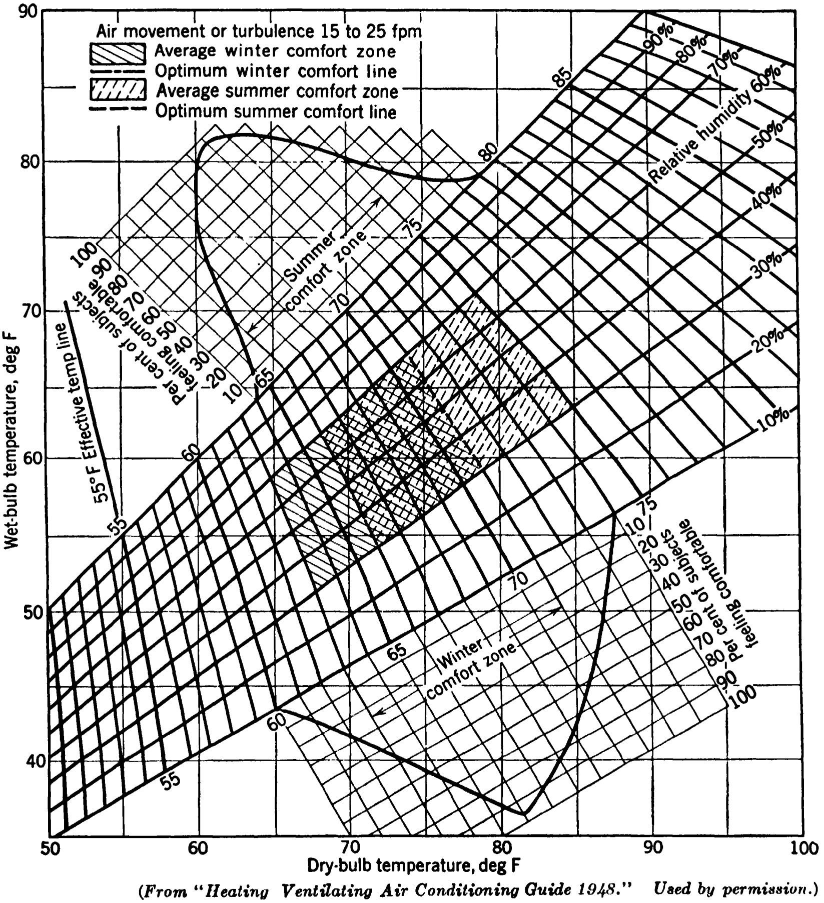
[1] "Determination of the Comfort Zone," by F. C. Houghten and C. P. Yaglou, *ASHVE Trans.*, Vol. 29, 1923.

"The Summer Comfort Zone: Climate and Clothing," by C. P. Yaglou and Philip Drinker, *ASHVE Trans.*, Vol. 35, 1929.

"How to Use the Effective Temperature Index and Comfort Charts," *ASHVE Trans.*, Vol. 38, 1932.

66 F for most people. For summer conditions the chart indicates an effective temperature of 71 F as the one desired by most people in the latitude of Pittsburgh, Pa. (approximately 40.5° N), and extending into southern Canada for cities not more than 1000 ft above sea level. Because of climatic conditions effective temperatures of 73 to 74 F may be desirable in the southern part of the United States. This indicates an increase of about a degree of effective temperature for each 5 deg of latitude decrease.



(From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.)

Fig. 18. ASHVE Comfort Chart for still air (velocities of 15 to 25 fpm).

Example. A room is to be maintained with a relative humidity of 30 per cent. What dry-bulb temperature is necessary to give an effective temperature of 66 F?

Solution. Beneath the intersection of the 66 F effective temperature curve and the 30 per cent relative humidity curve a dry-bulb temperature of 72 F is found on the axis of the abscissa.

Example. Find the required dry-bulb temperature if a relative humidity of 50 per cent is used with an effective temperature of 66 F.

Solution. Below the intersection of the curves of 66 F effective temperature and 50 per cent relative humidity find 70 F the dry-bulb temperature.

The range of relative humidities shown in the comfort zone is from 30 to 70 per cent. By the methods of the previous examples it may be shown that with 70 per cent relative humidity 68 F dbt will produce 66 F *ET*. Although regulation of the dry-bulb temperature will produce the same effective temperature with either a high or a low relative humidity, it is not always the case that, through the range of conditions indicated by a constant effective-temperature line, comfort effects exist. Either extremely high or low relative humidities may produce discomfort regardless of the existent effective temperature. The comfort chart clearly indicates that the optimum effective temperature of 66 F for winter is lower than that which may be required during warm weather.

The comfort chart, Fig. 18, is limited to use in the United States with certain restrictions. The winter comfort line applies where convection heating with a central plant is used; it does not apply where radiant heating alone is used and in spaces having very cold surrounding walls and excessive amounts of exposed glass areas. The summer comfort line is applicable to homes and offices where the occupants have been adjusted to the artificial conditions maintained. This latter line is not suitable for stores, theaters, etc., where the periods of exposure are less than three hours. The data of Fig. 18 are for adults who are at rest, or only slightly active, and are normally clothed for the season under consideration. Increased muscular activity produces conditions not comparable with those of the chart.

Air and water-vapor mixtures may have many (desirable or undesirable) combinations of psychrometric temperatures and velocities of movement in heating and air-conditioning systems. Figure 19,[2] based on laboratory investigations, indicates effective temperatures that may result with various temperature and velocity combinations for air-vapor mixtures. Attention should be given to the note in connection with Fig. 19 in regard to conditions for which the data are true.

Use is made of the graph by joining dry- and wet-bulb temperatures, located on their respective scales, by a straight line *AB* or any other straight line which may join a dry- and a wet-bulb temperature line. The wet-bulb temperature is never greater than the corresponding

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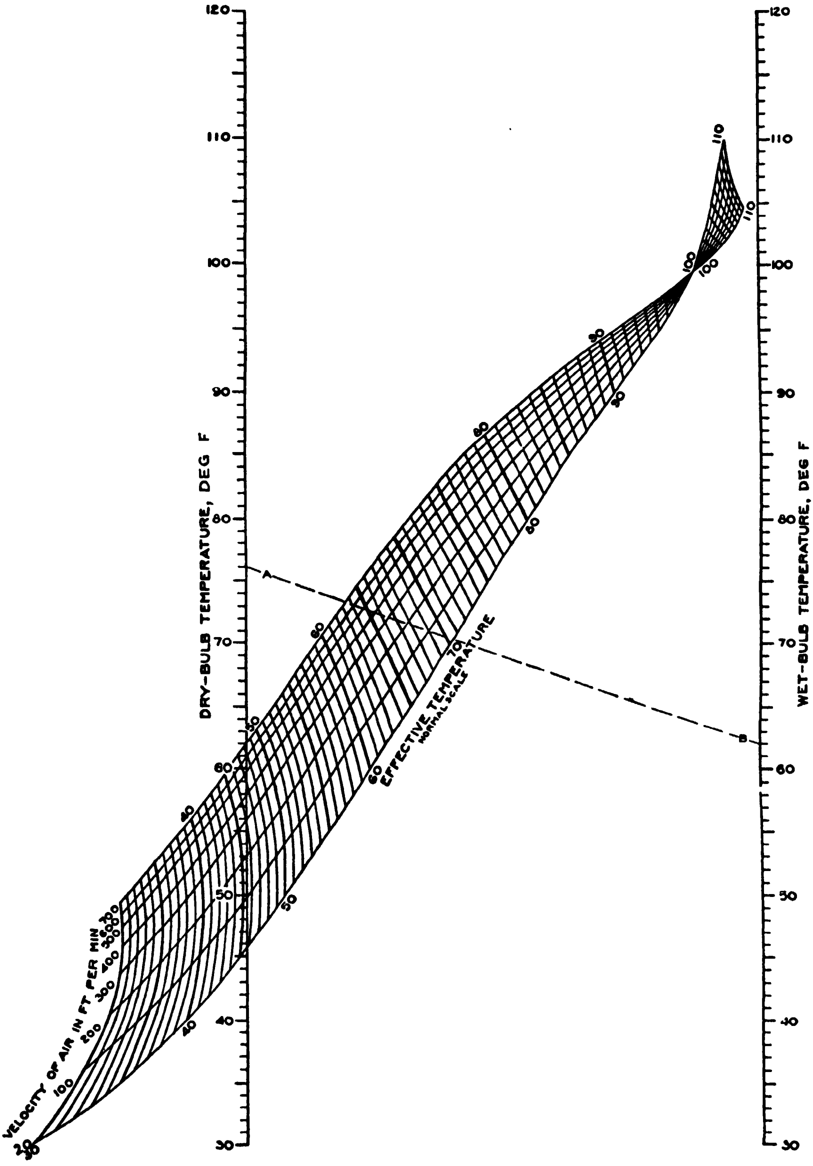
[2] "Effective Temperature with Clothing," by C. P. Yaglou and W. E. Miller, *ASHVE Trans.*, Vol. 31, 1925.



(*From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.*)

Fig. 19. Thermometric or effective temperature chart.

Applicable to inhabitants of the United States engaged in light muscular or sedentary work and wearing the customary indoor clothing where warm-air, direct steam, hot-water, or plenum heating systems are used.

dry-bulb reading. The effective temperature existent is located where the air-velocity curve cuts the line *AB*.

Based on still-air conditions (velocity 20 fpm) the data of Fig. 19 indicate for the following dry-bulb temperatures: (1) below 99.5 F either saturated or unsaturated air has its effective temperature decreased as the velocity of flow is made greater; (2) for temperatures ranging from 99.5 to 120 F the effective temperature of the mixture can be reduced by increases of air velocity, provided the wet-bulb temperatures are low enough; (3) saturated air at 99.5 F (slightly above the 98.6 F normal for human blood) has a constant effective temperature irrespective of its velocity of flow; (4) for each temperature ranging from 99.5 to 120 F together with one particular corresponding wet-bulb temperature somewhere between 99.5 and 96 F the effective temperature remains constant even though the velocity of flow is increased; and (5) for any combination of dry- and wet-bulb temperatures when both are above 99.5 F the effective temperature is made greater by increases of velocity of flow.

The human body cannot lose heat from its surfaces, by the process of convection, when its blood temperature is less than the dry-bulb temperature of the air flowing over it. The body surface areas may receive heat from the air by convection so that losses of heat by evaporation from and radiation from skin areas may be equalled or at least partially nullified. Such actions may result in no change of effective temperature or an increase in its values with a greater rate of air motion. For the major portion of the cases involved in air conditioning for comfort the effects of increased air motion are reflected by reduced effective temperatures.

60. Heat Production and Regulation in Man. The human organism is a form of heat engine which derives its energy from the combustion of fuel (food) within the body. This action, termed metabolism, is the process whereby the body produces heat and energy as the result of the oxidation of products within it by oxygen obtained from inhaled air. The rate of heat production is dependent upon the individual's health, his physical activities, and his environment. The normal blood temperature for most people in good health is 98.6 F, which is generally well above the temperature of the surrounding air. The body temperature maintained is dependent upon the heat generated within it and the heat lost from it by the processes of radiation, convection, and evaporation, either singly or in combination. The human organism is capable of some self-adaptation to the surrounding conditions, but its very sensitive methods of heat regulation are limited in the maintenance of heat equilibrium over a wide external temperature range. Therefore not only the amount of clothing worn by the individual but also his environment are important factors in the loss of heat from his body which is necessary for health and comfort. As long as the temperature of the surrounding air is below that of the blood the rate of air motion has a marked effect upon the losses of heat from the body by the process of convection.

In effecting loss of heat the body may react to bring more blood to the capillaries in the skin whereby heat losses may take place by radiation, convection, and some evaporation. When either the process of radiation or convection, or both, fails to produce the necessary loss of heat the sweat glands become more active, and more moisture is deposited upon the skin, carrying heat away as it evaporates. As long as the surrounding air and objects are below blood temperature, heat may be removed by the methods of radiation and convection. When the surrounding air is above the blood temperature the process becomes one of heat removal by evaporation only. If the body fails to throw off the requisite amount of heat the blood temperature rises. Consequently the human individual cannot safely exist for any considerable period of time in an atmosphere which is saturated and which together with surrounding objects is above his blood temperature, as heat cannot be lost from the body under those conditions.

The human body attempts to maintain its temperature when exposed to cold by the withdrawal of blood from the outer portions of the flesh, by decreased blood circulation, and by an increased rate of metabolism.

61. Heat and Moisture Losses from the Human Body. Heat is given off from the human body as either sensible or latent heat, or both. In order to design successfully any air-conditioning system for spaces which human bodies are to occupy, it is necessary to know the rates at which the two forms of heat are given off under different conditions of air temperature and bodily activity.

Figures 20, 21, 22, and 23, which were developed as results of research at the ASHVE laboratory[3] and which are reproduced by permission, supply information relative to total heat, sensible heat, and latent heat, respectively. Figure 20 is based on effective temperatures; Figs. 21 and 22 employ dry-bulb temperatures. Attention is called to the practically uniform conditions of heat losses over a considerable range of effective temperature for the various rates of physical activity as shown by Fig. 20. Figure 22 shows the grains of moisture given off per hour by an individual when engaged in different rates of physical activity. The curves of Fig. 23 are also of interest as they indicate the percentages of sensible and latent heat as given off at different dry-bulb air temperatures and various rates of work. Note that some of the curves indicate that all the heat is given off by evaporation even though the surrounding air temperature is less than 98.6 F.

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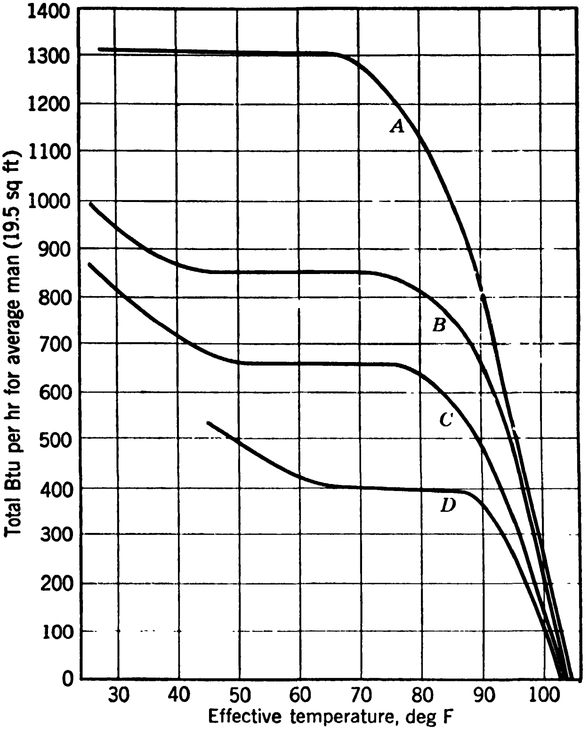
[3] "Heat and Moisture Losses from Men at Work and Application to Air-conditioning Problems," by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant, *ASHVE Trans.*, Vol. 37, 1931.



(*Copyright American Society of Heating and Ventilating Engineers, ASHVE Trans., Vol. 37, 1931.*)

FIG. 20. Relation between total heat loss from the human body and effective temperature for still air.

Curve *A*, men working 66,150 ft-lb per hr, metabolic rate 1310 Btu per hr. Curve *B*, men working 33,075 ft-lb per hr, metabolic rate 850 Btu per hr. Curve *C*, men working 16,538 ft-lb per hr, metabolic rate 660 Btu per hr. Curve *D*, men seated at rest, metabolic rate 400 Btu per hr. Curves *B* and *D* are based on test data which cover a wide temperature range. Curves *A* and *C* are based on test data at an effective temperature of 70 F and extrapolation of curves *B* and *D* which are from data at many temperatures. All curves are averages of data for high and low relative humidities; variations due to humidity are small.

Curve *D* of Fig. 22 shows that, at a dry-bulb air temperature of 80 F, 1180 grains of moisture and 175 Btu of latent heat per hr will be given off by a man at rest. The latent heat at blood temperature of 98.6 F may be found in Table 6, Chap. 1, as 1038 Btu per lb of vapor. The moisture represented by 1180 grains is 1180 ÷ 7000 = 0.1685 lb. The calculated latent heat for the conditions stated is 1038 × 0.1685 = 174.9 Btu per hr, which checks the value of 175 Btu per hr as read from the chart.

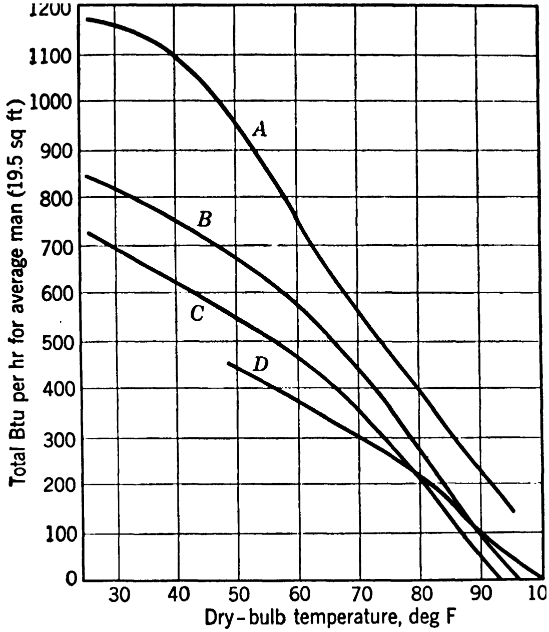


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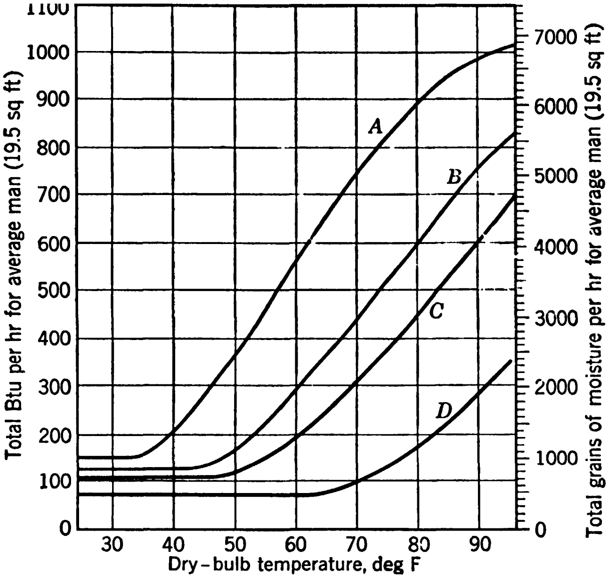
FIG. 21. Relation between sensible heat loss from the human body and dry-bulb temperature for still air.

Curve *A*, men working 66,150 ft-lb per hr, metabolic rate 1310 Btu per hr. Curve *B*, men working 33,075 ft-lb per hr, metabolic rate 850 Btu per hr. Curve *C*, men working 16,538 ft-lb per hr, metabolic rate 660 Btu per hr. Curve *D*, men seated at rest, metabolic rate 400 Btu per hr. Curves *B* and *D* are based on test data which cover a wide temperature range. Curves *A* and *C* are based on test data at an effective temperature of 70 F and extrapolation of curves *B* and *D* which are from data at many temperatures. All curves are averages of data for high and low relative humidities; variations due to humidity are small.

62. Moisture Content of Air. The discussion of effective temperature brought out the fact that air dry-bulb temperature, relative humidity, and air motion are interrelated. Once two of the foregoing items are fixed, the third must be whatever is required to give the necessary effective temperature to produce comfort conditions.

Owing to weather conditions the moisture content of outside air may be low during cold weather and above the average during hot weather as the capacity of the air to carry moisture is dependent upon its dry-bulb temperature. This means that, in the winter, inleakage of cold outside air, having a low-moisture content, will cause a low relative humidity in heated spaces unless moisture is added to the air by the process of humidification. In the summer the reverse is likely to occur unless moisture is removed from the inside air by a dehumidification process. This is especially true if the space is densely occupied by people, if sources of moisture such as steam tables and coffee urns in restaurants are present, and if the outside air is humid.



(Copyright American Society of Heating and Ventilating Engineers. ASHVE Trans., Vol. 37, 1931.)

FIG. 22. Latent heat and moisture loss from the human body by evaporation in relation to dry-bulb temperature for still-air conditions.

Curve *A*, men working 66,150 ft-lb per hr, metabolic rate 1310 Btu per hr. Curve *B*, men working 33,075 ft-lb per hr, metabolic rate 850 Btu per hr. Curve *C*, men working 16,538 ft-lb per hr, metabolic rate 660 Btu per hr. Curve *D*, men seated at rest, metabolic rate 400 Btu per hr. Curves *B* and *D* are based on test data which cover a wide temperature range. Curves *A* and *C* are based on test data at an effective temperature of 70 F and extrapolation of curves *B* and *D* which are from data at many temperatures. All curves are averages of data for high and low relative humidities; variations due to humidity are small.

For economic reasons the inside air temperature of heated structures should be carried as low as is possible and that of cooled structures as high as is possible and still produce conditions of comfort. For people normally clothed and slightly active the usual inside specified air temperature at the breathing line (5 ft above the floor) is about 70 F in the winter time and somewhere around 80 F in the summer.

A deficient amount of moisture in air maintained at 70 F, in the winter, will result in more rapid evaporation of moisture from the individual's exposed skin surfaces so that it may be necessary to maintain a dry-bulb temperature of the air anywhere from 71 to 75 F to produce a sensation of comfort for the individual. The higher inside air temperature thus required results in increased operating costs of the heating plant, and the further decreased relative humidity of the air may cause injury to woodwork, furniture, etc., and to the surfaces of the nasal passages of the occupants. Excessive moisture in the air, in either summer or winter, will result in a condition of discomfort at a given dry-bulb temperature owing to the fact that heat losses from the human body by the process of evaporation are materially

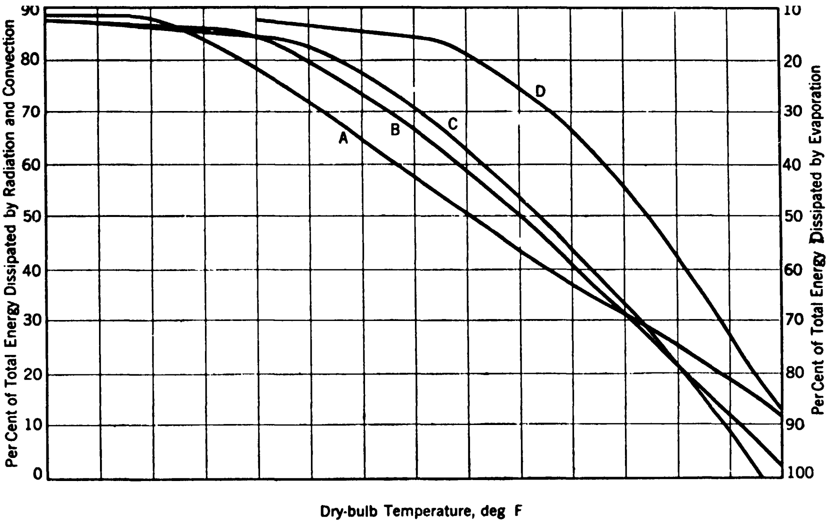


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FIG. 23. Heat loss from the human body by evaporation, radiation, and convection in relation to dry-bulb temperature for still-air conditions.

Curve *A*, men working 66,150 ft-lb per hr, metabolic rate 1310 Btu per hr. Curve *B*, men working 33,075 ft-lb per hr, metabolic rate 850 Btu per hr. Curve *C*, men working 16,538 ft-lb per hr, metabolic rate 660 Btu per hr. Curve *D*, men seated at rest, metabolic rate 400 Btu per hr. Curves *B* and *D* are based on test data which cover a wide temperature range. Curves *A* and *C* are based on test data at an effective temperature of 70 F and extrapolation of curves *B* and *D* which are from data at many temperatures. All curves are averages of data for high and low relative humidities; variations due to humidity are small.

reduced. In summer cooling, the moisture content of the air must be reduced to the proper value for comfort by the process of dehumidification. In selecting the proper dry-bulb air temperature for either summer or winter conditions the designer must be influenced by the practical consideration of relative humidities which are feasible. In general for winter conditions, in the average residence, relative humidities above 35 to 40 per cent are not practical. In summer comfort cooling the air of the occupied space should not have a relative humidity above 50 per cent. With these limitations as to relative humidity the designer may determine from the Comfort Chart, Fig. 18, the necessary dry-bulb temperatures for the air. In industrial air conditioning where relative humidities either higher or lower than those mentioned for residence work are imperative it will be necessary to design and equip the structure so that the required relative humidities are feasible and practical.

63. Air Motion. In all systems of heating, except radiant heating, and in all systems of summer cooling the air of the space conditioned ultimately becomes a carrier of heat. In heating, the air gives up heat; in cooling, it absorbs heat. Therefore if uniform conditions are to be obtained the air must be properly distributed throughout the spaces served. This is necessary in order that the heat either be properly supplied or removed, and it is also necessary to prevent the concentration of moisture, odors, toxic gases, etc., in various locations. No system of heating, ventilating and air conditioning is either satisfactory or successful unless the air handled is properly circulated and distributed.

The velocity of the air in the occupied zone is important as it must never be such as to produce an objectionable draft. The air velocity in the occupied zone ordinarily should not exceed 25 to 40 fpm in summer and winter installations and should be just enough to give a pleasant feeling. The air velocities in the space above the occupied zone may be anything which is necessary to produce good distribution of the air in the space, provided of course that the air in motion does not produce an objectionable noise. Whenever possible the flow of air should be toward the faces of the individuals in the occupied zone rather than from the rear.

64. Cold and Hot Surfaces. Discomfort may be occasioned by the presence of either cold or hot objects in a space. In winter heating, the flow of heat is outward from the air of a heated space to the outside air. Areas where the transmission of heat occurs are walls, windows, doors, ceilings, floors, etc. Irrespective of the building construction the inside-surface temperatures of such areas will always be less than that of the air inside the room when an outward flow of heat takes place. The amount of lowering of the inside-surface temperature is dependent upon the outside-air temperature, the inside-air temperature, and the resistance which the wall, window glass, etc., offers to the flow of heat. Areas through which heat flows rapidly will have a lower inside-surface temperature than those having a lesser rate of heat flow for the same inside-outside air-temperature differential. Cold surfaces at the insides of walls and windows receive heat from the bodies of room occupants by the process of radiation. This loss of heat from the human body often causes a sensation of chilliness and discomfort. The inside-surface temperatures of walls may be increased by the use of insulation to reduce the rate of heat transmission. The inside-surface temperatures at glass areas, such as windows, can be increased by the use of two or three panes of glass, separated from each other by air spaces, to reduce the amount of heat lost through the glass. Storm sashes properly fitted to windows are effective in reducing the loss of heat through such areas and also in preventing the inleakage of cold air which often produces objectionable drafts as well as an increased heating load. Walls and windows which are resistant to the flow of heat are also of benefit in summer cooling installations in relieving the plant of load and in the production of more comfortable conditions in the air-cooled space. Hot objects such as industrial furnaces and heated materials which are being either processed or fabricated are sources of heat.

Radiant heat from various sources may be received by the surfaces of the human body to produce a sensation of heat, and furthermore heated objects may also liberate heat by the process of convection to produce unsatisfactory air temperatures. Therefore the temperatures of surfaces to which the body may be exposed are of considerable importance in the design of heating, ventilating, and air-conditioning systems. A discussion of heat transfer by radiation is given in Art. 17. Material relative to the control of body surface temperatures, through the agency of radiation processes, is included in Chap. 12, Panel Heating.

65. Air Stratification. Air when heated becomes less dense and rises to the upper part of its confining envelop when subject to the action of gravity alone. This action occurs in the heated spaces of structures no matter what type of heating system is employed. The result is that there may be a considerable variation in the air temperatures between the floor and the ceiling levels. The movement of the air to produce the temperature gradient from floor to ceiling is termed air stratification. Certain types of heating systems and certain arrangements of heating systems do more to reduce the stratification of air than others. The reduction of the stratification of air in a space is important from two standpoints: (1) reduction of the heat losses from the upper portion of the room, and (2) the maintenance of comfortable air temperatures in the occupied zone.

Generally heating specifications call for the maintenance of a definite air temperature at the breathing-line level, which is 5 ft above the floor. When an adult is standing most of his body is below the 5-ft level; when he is seated all of it is below that level. Therefore if a large temperature gradient exists between the floor and the 5-ft level the body is in an atmosphere which may be too cool for comfort. Consideration should always be given to the problem of reducing air temperature gradients to a minimum, and, when necessary, the inside air temperature desired should be specified at the 30-in. level, which is midway between the floor and the so called breathing-line level.

66. Appraisal of Comfort Conditions. An evaluation of comfort conditions should include a study of the effects of radiation, air dry-bulb temperatures, relative humidities, and air motion. The ASHVE Comfort Chart, Fig. 18, is based on the last three items enumerated, the data for which may be obtained by the use of a suitable hygrometer or psychrometer and a means of determination of air velocities. For low velocities of air flow either the Kata thermometer, Art. 278, or a hot-wire anemometer may be used. The hot-wire anemometer has a fine bare resistance-wire element maintained at a constant temperature above that of the air by means of measured electrical current. The amount of current flowing varies as the rate of air flow changes and from calibration data the velocity of the air may be ascertained.

When thermal interchanges take place the following instruments[4] are of use in the appraisal of conditions as they affect the comfort of the human body: globe thermometer, eupatheoscope, thermo-integrator, heated globe, and the Kata thermometer.

A globe-thermometer unit consists of an ordinary mercurial thermometer, Fig. 24, placed so that its bulb is at the center of a blackened hollow sphere which has a diameter of 8 in. The thermometer receives radiant heat from all directions when the instrument is properly suspended and provides a means of measuring radiant effects. Inasmuch as the human form is not a sphere the instrument has limitations in measuring the effects of radiant heat upon the body surfaces. Consequently the direction from which the radiations come should be taken into account as the projected areas of the human form vary considerably.

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[4] "Measurement of the Physical Properties of the Thermal Environment," by D. W. Nelson, F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, A. P. Gagge, John A. Goff, A. E. Hershey, F. C. McIntosh, F. W. Reichelderfer, G. L. Tuve, and C. P. Yaglou, *ASHVE Journal Section of Heating, Piping and Air Conditioning,* June, 1942.

When the radiant surfaces are warmer than the ambient air surrounding the instrument the temperature measured by the thermometer within the globe is greater than that of the surrounding air as indicated by a properly supported and shielded thermometer. The reverse is true when the surrounding room-wall and other surface areas are cooler than the air of the space. A globe thermometer comes into equilibrium with its surroundings rather quickly, and the effects[5] of radiation and convection can be shown to balance each other. Air velocities existent in the vicinity of a globe thermometer can be measured by means of either a Kata thermometer or a hot-wire anemometer. Calibration of a globe thermometer involves data pertinent to the readings of the internal thermometer, the room air temperature, and the velocity of air flow. From these data the mean radiant temperature existent can be established by means of some form of a graph or chart.

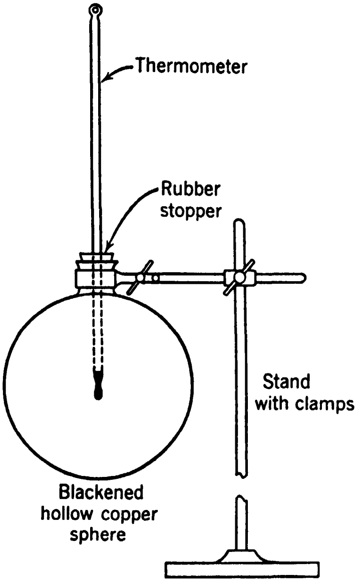

FIG. 24. Globe thermometer.

The eupatheoscope is a device designed to measure the effects of radiation upon the human body. The instrument consists of a blackened hollow copper cylinder of $7\frac{1}{2}$-in. diameter and 22-in. height, which gives the proper surface area. The cylinder surface is maintained at 80 F (the approximate mean surface temperature of the human body) by two thermostatically controlled electric bulbs. A portion of the lamp current is passed through a coil wound around a mercury thermometer which furnishes a measure of the heat necessary to maintain the surface temperature of the cylinder at the required value. The thermometer indicates the so-called equivalent temperature. This instrument has been used in connection with studies of the performance of direct steam radiators, as a control in radiant-heating systems, and as a measure of cooling effects in occupied spaces. The device has limitations when the room air is warmer than the desired surface temperature and when it is too cool for the lamps to maintain the proper surface temperature.

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[5] "The Globe Thermometer in Studies of Heating and Ventilating, by Bedford and Warner," *Journal of Hygiene*, Vol. 34, 1934.

The thermo-integrator[6] consists of an electroplated hollow copper cylinder made of metal about 0.05 in. thick. The cylinder has a diameter of 8 in., a length of 24 in., and hemispherical ends. Each end has, located on the longitudinal axis of the device, an externally threaded nipple with an internal bore of $\frac{1}{2}$ in. and a length of approximately one inch. Each nipple has a threaded cap with a central opening. One brass cap has a $\frac{1}{8}$-in. round brass rod, and the other cap has a $\frac{1}{4}$-in. copper tube passing through it into the cylinder. The rod and the tube are insulated from the caps, in each case by means of a rubber stopper, through which they pass. A suitable length of No. 24 nichrome wire joins the inside end of the rod with that of the tube and serves as an internal heater for the instrument. Air stratification and convection movements within the instrument are minimized by evacuating the cylinder through the copper tube until the internal air pressure is less than 0.1 per cent of an atmosphere. The heater circuit embodies two 6-volt storage batteries, and adjustments are made to deliver as uniformly as possible 21.5 watts of energy to the heater. This arrangement permits the instrument to dissipate 17.5 Btu per hr per sq ft of surface. The average surface temperature is measured by eight iron-constantan thermocouples placed about the surface of the cylinder and connected in parallel to an automatic recorder.

In use the thermo-integrator is operated with a constant current of 2.5 amperes with frequent checks of the voltage drop across the heater to insure the proper wattage input. Temperatures of the ambient air and the surfaces of the cylinder are measured after equilibrium conditions have been established for the instrument in the space under consideration. The surface temperature of the thermo-integrator is affected by radiation together with room-air temperature and movement. The effects of the relative humidity of the room air are not included. Relative humidity has an effect on comfort conditions. However, greater changes of relative humidity may be required to affect comfort conditions that are resultant from the effects of changes of the amount of radiant energy involved.

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[6] "The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges," by C.-E. A. Winslow and Leonard Greenburg, *ASHVE Trans.*, Vol. 41, 1935.

The thermo-integrator sums up all radiation effects in a room. When the instrument is properly calibrated it can be used to measure them by subtracting from its measured total-heat losses those occasioned by the room-air temperature and velocity.

The Kata thermometer may be used as a wet-bulb instrument in the study of cooling effects in the appraisal of comfort conditions. This device has limitations in its use.

PROBLEMS

1. What is the effective temperature of still air having dry- and wet-bulb temperatures of 70 and 58 F? Are conditions favorable for winter comfort?

2. Air is at 72 F dry-bulb temperature and 40 per cent relative humidity. What is the effective temperature if the velocity is 20 fpm?

3. Still air under summer conditions has a dry-bulb temperature of 83 F and 50 per cent relative humidity. What is the effective temperature?

4. Air moves with a velocity of 200 fpm when its dry- and wet-bulb temperatures are 93 and 57 F. What is the effective temperature?

5. What is the effective temperature for the following conditions: air velocity, 100 fpm; dry-bulb temperature, 103 F; and wet-bulb temperature, 55 F?

6. Air moving at a velocity of 20 fpm has a dry-bulb temperature of 120 F and a wet-bulb temperature of 100 F. What is the effective temperature of the air?

7. Find the effective temperatures for air at 115 dbt and 103 F wbt for velocities of 20 fpm and 300 fpm.

8. What total heat will be given off, Btu per hour, by a man working at the rate of 33,075 ft-lb per hr in an atmosphere having an effective temperature of 84 F?

9. What total sensible heat is given off, in Btu per hour, by an individual at rest in an atmosphere having a dry-bulb temperature of 80 F? What amount of latent heat and how many grains of moisture are given off per hour under the same conditions?

10. Find the sensible heat, the latent heat, and the moisture given off per hour by an individual working at the rate of 16,538 ft-lb per hr when in an atmosphere having a dry-bulb temperature of 70 F.

11. What percentage of the heat given off by an individual at rest in still air having a dry-bulb temperature of 85 F is sensible heat? latent heat?

12. At what dry-bulb temperature will 90 per cent of the heat given off by an individual working at the rate of 33,075 ft-lb per hr be by evaporation?

CHAPTER 4

HEAT TRANSMISSION AND HEAT LOSSES FROM BUILDINGS

67. Foreword. No building is impervious to the flow of heat through its enclosing walls, windows, floors, ceilings, etc. The same thing is true also in regard to the inleakage of outside air (infiltration) and the outward leakage of inside air (exfiltration) through porous walls and through cracks in the walls and about window and door openings.

Normally the flow of heat is from a region of higher to one of lower temperature. Therefore, during the winter when the outside air temperatures are lower than those inside, a structure suffers losses of heat. The converse is true in the summer when cooling of a building may be necessary to maintain the inside temperatures at a lower value than that of the outside air, so that the building has a gain of heat. Whether the load be that of summer cooling or winter heating, the items of heat transmission through the building materials and air infiltration are large factors of the total plant load to be either supplied or abstracted.

Each structure presents an independent problem with its materials of construction, design, location, and care and workmanship of construction. Every structure is an integral part of the plant which serves it, and the performance of either a heating or a cooling plant is influenced by the individual characteristics of the building served.

68. Heat Transmission of Building Materials. In Arts. 15, 16, and 17, it was shown that heat may be disseminated by conduction, convection, and radiation. The transfer may take place by any of the foregoing methods, singly or collectively. In respect to building materials the transmission of heat is affected by the following items: (1) their character and thickness, (2) their arrangement in the building construction, (3) the temperature differences maintained, (4) their mean absolute temperatures, (5) the movement of air over exposed surfaces, and (6) the time interval during which the flow of heat occurs.

69. Overall Coefficient of Heat Transmission. The calculation of the heat transmitted by either conduction, convection, or radiation involves the use of surface temperatures, which are often difficult to obtain. Therefore a coefficient of heat transmission is desirable which takes into account the effects of conduction, convection, and radiation together with the kind, thickness, and placement of the materials and which may be used with the difference of the air temperatures existent adjacent to each side of either the wall or other section under consideration. Such a numerical quantity is the overall coefficient of heat transmission, U, which is defined as the amount of heat, in Btu per hour, transmitted per square foot of area of the existent section per one degree Fahrenheit of inside- and outside-air-temperature difference.

In equation form, the expression of the transmission of heat through each component portion of the enclosing envelope of a space is

$$H = UA(t - t_o) \tag{40}$$

where H = heat transmitted per hour, Btu.
U = overall coefficient of heat transmission.
A = area of wall, floor, ceiling, roof, or glass, sq ft.
t = inside air temperature, deg F.
t_o = outside air temperature, deg F.

The overall coefficient of heat transmission for a wall section may be determined either experimentally or by calculations involving the use of known data for the materials included. The experimental determination of overall coefficients is both laborious and expensive for all the building materials and the combinations in which they may be used in walls. Experimental determinations of U are used largely to check the accuracy of values determined by calculation as the necessary data for the calculations can be found with a smaller amount of test work.

70. Experimental Determination of Overall Coefficients of Heat Transmission. The commonly used method of determination of the coefficient U is that of the guarded hot box,[1] Fig. 25. The apparatus consists of an outer box made of five insulated walls each containing 3 in. of corkboard or its equivalent and of 5 ft × 5 ft inside dimensions. Within the outer box is an inner box with five insulated walls each containing 2 in. of corkboard or its equivalent and of 3 ft × 3 ft inside dimensions. The open sides of the two boxes are placed in the same plane with uniform spacings between all the other corresponding walls of the inner and outer boxes. The wall test specimen is clamped tightly against the edges of the inner and outer boxes at their common open side, the joints between the wall edges and the test specimen being made tight by means of gaskets. The same uniform air temperature conditions are maintained by the use of non-luminous electrical heaters and motor-driven fans both within the inner box and the air space between the inner and the outer boxes. The electrical input to the fan and the heater unit of the inner box, reduced to Btu, is a measure of the heat dissipated through an area of A of the test wall section which is exposed to the interior of the inner box. The purpose of the fans in connection with the electrical heaters in all air spaces is to give a gentle air motion to prevent great variations in air temperature due to stratification in the spaces. Measurements to secure the average air temperature t within the inner box are necessary. The same is true in regard to the temperature of the air t_o in contact with the outer surface of the test specimen. The overall coefficient of heat transmission is calculated from the test data as

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[1] "Standard Code for Heat Transmission through Walls," *ASHVE Trans.*, Vol. 34, 1928.

$$U = \frac{\text{Watt-hours input per hour to inner box} \times 3.412}{A(t - t_o)} \tag{41}$$

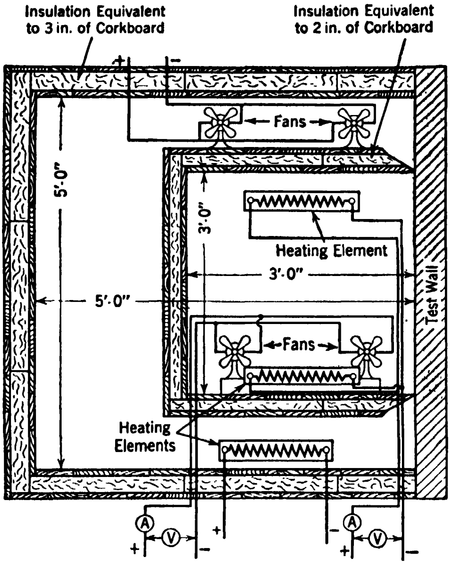
Insulation Equivalent to 3 in. of Corkboard
Insulation Equivalent to 2 in. of Corkboard
+
-
Fans
5'-0"
3'-0"
Heating Element
3'-0"
5'-0"
Test Wall
Fans
Heating Elements
A
V
+
-
+
-
A
V
+
-

FIG. 25. Guarded hot box.

The Nicholls heat meter[2] has been developed for use in the determination of the overall coefficients of heat transmission of actual building walls in place and subjected to actual weather conditions. The apparatus consists of a piece of Bakelite $\frac{1}{8}$ in. thick and 2 ft square fitted with a number of commonly connected thermocouples embedded flush with its surfaces. The thermocouples measure the difference in electrical potential caused by the temperature differential between the two surfaces of the plate when it is firmly clamped to the test wall. The meter is calibrated in terms of the hourly quantities of heat, H_t, passing through the plate when various electrical potential differentials exist. When the inside and outside air temperatures are known, $U = H_t \div A(t - t_o)$.

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[2] "Measuring Heat Transmission in Building Structures and a Heat Transmission Meter," by P. Nicholls, *ASHVE Trans.*, Vol. 30, 1924.

71. Calculation of *U*. Heat passing through a building wall composed of a single homogeneous material, as in Fig. 26, is received at the wall surface exposed to the region of higher air temperature by two methods, radiation and convection. The flow then takes place by conduction through the material to the surface exposed at the region of lower air temperature where the heat is dispersed through the processes of radiation and convection.

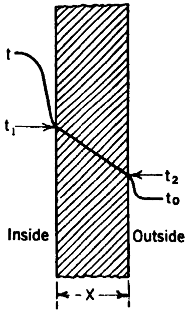

Fig. 26. Simple wall.

Figure 26 represents a case where the inside air temperature t of a room is greater than t_o the outside air temperature. The inside wall-surface temperature is represented by t_1 and the outside wall-surface temperature by t_2. The temperatures drop in numerical value from t to t_o, and the rates of drop are dependent upon the resistances which are offered to heat flow.

The amount of heat H entering the wall per hour per unit of area is fixed by the temperature drop $t - t_1$ and the combined coefficient of radiation and convection f_i which is effective at the warmer surface of the wall. The hourly quantity of heat H received per unit area (1 square foot) of the wall is $f_i(t - t_1)$. The foregoing quantity of heat then flows by conduction through the material from the surface at a temperature of t_1 to the surface having the lower temperature of t_2. The heat transfer by conduction, Art. 15, is $H = (k/x)(t_1 - t_2)$ Btu per hr per sq ft. The symbol x designates the material thickness in inches. k is the thermal conductivity; it is the amount of heat in Btu per hour passing by conduction through a section of the material one inch in thickness and one square foot in area, per one degree Fahrenheit difference between the temperatures of the surfaces. Finally the loss of heat from the surface having the lower temperature of t_2 is $H = f_o(t_2 - t_o)$ where f_o is the combined effect of radiation and convection for the conditions which prevail at the wall surface of lower temperature. The items f_o and f_i, designated as surface conductances, are expressed as Btu per hour, per square foot of surface, per degree Fahrenheit of temperature difference and are markedly affected both by air movement and the nature of the wall surface. When the wall surface conditions and the air movement are different at the two surfaces, f_i and f_o are not equal, and $(t - t_1)$ and $(t_2 - t_o)$ cannot be equal.

As the same quantity of heat is involved in each of the examples of transmission cited for Fig. 26 and is also equal to $U(t - t_o)$, then

$$H = f_i(t - t_1) = \frac{k}{x}(t_1 - t_2) = f_o(t_2 - t_o) = U(t - t_o)$$

and

$$\frac{U}{f_i}(t - t_o) = (t - t_1) \qquad \frac{Ux}{k}(t - t_o) = (t_1 - t_2)$$

and

$$\frac{U}{f_o}(t - t_o) = (t_2 - t_o)$$

Hence

$$\left(\frac{U}{f_i} + \frac{U}{f_o} + \frac{Ux}{k}\right)(t - t_o) = (t - t_1) + (t_2 - t_o) + (t_1 - t_2)$$

$$U\left(\frac{1}{f_i} + \frac{1}{f_o} + \frac{x}{k}\right)(t - t_o) = (t - t_o)$$

and

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x}{k}} \tag{42}$$

The fractions of the denominator of equation 42 are resistances to heat flow. The film or surface resistances are $1/f_i = R_i$ and $1/f_o = R_o$; the resistance to heat transfer by conduction $x/k = R_c$. The sum of R_i, R_o, and R_c is equal to R_t, the total resistance for a simple wall. U the overall coefficient of heat transmission is equal to the reciprocal of R_t and is

$$U = \frac{1}{R_t} = \frac{1}{R_i + R_o + R_c} \tag{43}$$

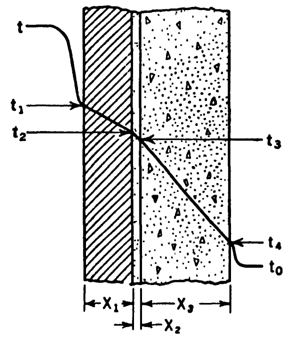

Fig. 27. Compound wall.

72. Transmission Coefficients for Compound Walls. When building walls are constructed of layers of different materials, Fig. 27, and when air spaces are placed between materials, as shown by Fig. 28, modifications of equation 42 are necessary. The equation changes must take into account the conductivities and thickness of the materials and the additional surfaces by which heat enters and leaves the materials. For the wall illustrated by Fig. 27, equation 42 would merely be extended to include the extra thicknesses of materials and is expressed

$$U = \frac{1}{R_t} = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x_1}{k_1} + \dfrac{x_2}{k_2} + \dfrac{x_3}{k_3}} \tag{44}$$

A conductance, *C*, is defined as the amount of heat in Btu per hour passing through one square foot of area of any material of the thickness and arrangement stated, per one degree Fahrenheit difference of the material surface temperatures. Note that a conductance is always written into the equation for *U* as $1/C$, and never otherwise. The term *a* is used to designate the conductances of air spaces of various widths and various air temperatures. Values for air-space conductances are given by Fig. 31.

The expression of *U* for the compound wall of Fig. 28, which includes several materials and an air space, is

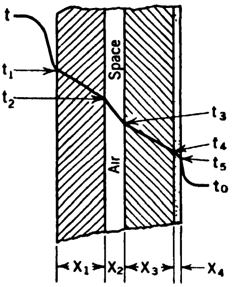

Fig. 28. Wall with air space.

$$U = \frac{1}{R_t} = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x_1}{k_1} + \dfrac{1}{a} + \dfrac{x_3}{k_3} + \dfrac{x_4}{k_4}} \tag{45}$$

Equation 45 may be modified to include any number of materials of different thickness and any number of and width of air spaces. A conductance of 1.10 is commonly applied where the width of an air space is greater than 2 in., as in Fig. 32 and Tables 13 and 14.

Typical walls of wood frame construction having air spaces are shown by Fig. 29.[3] Actual temperature gradients through the walls are given for both wind and still air conditions at their exteriors.

73. Surface or Film Conductances. The inside-surface conductance f_i and the outside-surface conductance f_o are affected by the surface material, its condition as to finish, the surface temperature, and the temperature of the adjacent air. With still air in contact with both the inner and outer surfaces and with all other conditions identical f_1 and f_o are equal for a given material.

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[3] "Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields," by A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo, *University of Illinois Engineering Experiment Station Bulletin* 192.

Air movement over the surface of a material very definitely increases the surface conductance. The rate of increase is nearly directly proportional to the increase of air velocity over a given material under fixed conditions. The rate of change of the surface conductance with air-velocity change varies with the material, being more for some materials than for others. Surface conductances for all materials at conditions of constant air velocity increase with an increase of mean temperature. The mean temperature is the average of the surface temperature and the temperature of the adjacent air which may be measured at a distance of one inch from the surface. Air temperature gradients may exist at a distance as far out as $1\frac{1}{2}$ in. from the surface with air velocities of one mile per hour and at a distance not to exceed one inch from the surface with air velocities exceeding 5 mph with the test surface mounted in the side of a duct wall. With air velocities in open spaces their gradients or variations are effective for distances of 10 to 13 in. from the surface for velocities up to 30 mph.

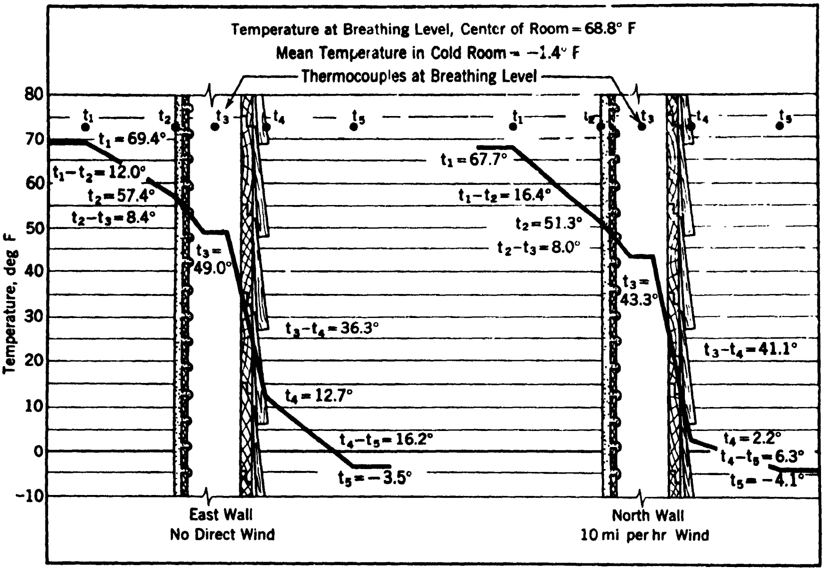

FIG. 29. Temperature gradients through walls of frame construction.

Under service conditions the air movement may be either parallel to the wall surface or at some angle to it. Angular incidence of the air at the surface tends to reduce the surface coefficient some for otherwise fixed conditions. However, the general practice is to assume air flow parallel to the surface.

Values of f_i vary from 1.4 to 2.1 for still-air conditions with different materials, surface conditions, and mean temperatures. The variation of the conductance f_o[4] for conditions of parallel air flow at several velocities, with different materials and surfaces all with a mean temperature of 20 F, is shown by Fig. 30. Equations developed by Houghton and McDermott indicate that f_o varies with the wind velocity, in v mph, as

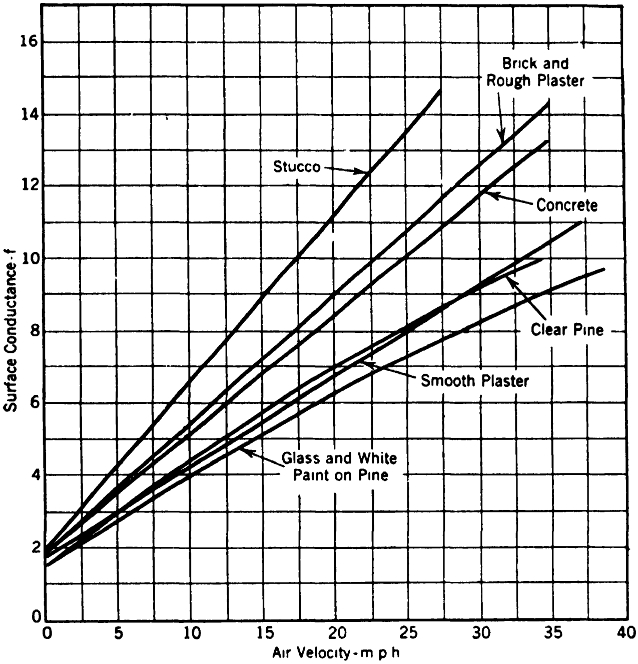

FIG. 30. Variation of surface conductances with wind velocity.

$$f_o = 1.4 + 0.28v \text{ for very smooth surfaces} \tag{46}$$

$$f_o = 1.6 + 0.3v \text{ for smooth walls and plaster} \tag{47}$$

$$f_o = 2.0 + 0.4v \text{ for cast concrete and smooth bricks} \tag{48}$$

$$f_o = 2.1 + 0.5v \text{ for rough stucco} \tag{49}$$

Small numerical changes of either f_i or f_o do not produce large changes in the calculated value of the overall coefficient U. Commonly

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[4] "Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface," by F. B. Rowley, A. B. Algren, and J. L. Blackshaw, ASHVE Cooperative Research Paper, 869, *ASHVE Trans.*, Vol. 36, 1930.

TABLE 10

THERMAL CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING MATERIALS

Material	Weight, Lb per Cu Ft	Mean Temp, Deg F	k	C	Investigator or Authority
Brick, common			5.00		*ASHVE Guide*
Brick, face			9.20		*ASHVE Guide*
Brickwork, wet			5.00		Willard, Lichty, and Harding
Concrete, average			12.00		*ASHVE Guide*
Concrete, stone, 1-2-4 mix	143	69	9.46		F. B. Rowley
Concrete, cinder	110	75	5.20		J. C. Peebles
Mortar, cement			8.00		Willard, Lichty, and Harding
Plaster		73	3.3		F. B. Rowley
Plaster, gypsum			5.0		*ASHVE Guide*
Plaster ⅜ in. thick with ⅜ in. wood lath. Total ¾ in.		70		2.5	F. B. Rowley
Roofing, composition		75		6.5	J. C. Peebles
Shingles, asbestos	65	75		6.00	J. C. Peebles
Shingles, asphalt	70	75		6.50	J. C. Peebles
Stone, average			12.5		*ASHVE Guide*
Stucco			12.0		*ASHVE Guide*
Tile, hollow clay 8 in. 12 in. 16 in.				0.60 0.40 0.31	*ASHVE Guide*
Tile, hollow clay with ½ in. cement plaster on each side 2 in. 4 in. 6 in.	120 127 124.3	110 100 105		1.00 0.60 0.47	Willard, Lichty, and Harding
Wood across the grain Cypress Fir Maple Yellow pine White pine	28.7 26.0 44.3 40.0 31.2	86 75 86 75 86	0.67 0.76 1.10 1.03 0.78		U. S. Bureau of Standards F. B. Rowley U. S. Bureau of Standards F. B. Rowley U. S. Bureau of Standards

accepted values for all materials are: f_i as 1.65 for still-air conditions and f_o as 6.0 for all materials when the average wind velocity over their surfaces is 15 mph. The foregoing average values of f_i and f_o were used in calculating the coefficients U given by Fig. 32 and Tables 12, 13, and 14.

74. Thermal Conductivities and Conductances. Definitions of the foregoing items have been presented in Arts. 71 and 72. The thermal conductivities, k, for building materials vary greatly and are dependent upon density, moisture content, age and proportions of the mix in the case of concrete, and the mean temperature. The last item is important as the conductivity increases with an increase of the temperature of the material. Therefore, the mean temperature should always be stated with the value of k for any material. In general, the materials having the smaller weights per cubic foot (density) have lower unit rates of heat transfer by conduction. The same is true with respect

TABLE 11

THERMAL CONDUCTIVITIES k AND CONDUCTANCES C FOR INSULATING MATERIALS

Material	Weight, Lb per Cu Ft	Mean Temp, Deg F	k	C
Air spaces, $\frac{3}{4}$ in. or more. Vertical bounded by ordinary material				1.10
Vertical bounded by aluminum foil				0.46
Horizontal, with surface emissivity 0.83				1.21
Asbestos and cement board	123	86	2.70	
Bark, redwood	5	75	0.26	
Corkboard	7	90	0.25	
	14	90	0.26	
Cork, regranulated	8.1	90	0.31	
Eel grass, between sheets of strong paper	3.4	90	0.25	
	4.6	90	0.26	
Flax fibers, between sheets of paper	4.9	90	0.28	
Glass wool	10	90	0.27	
Gypsum board, $\frac{1}{2}$ in. thick	53.5	90		2.60
Hair, felted	11	90	0.26	
	13	90	0.26	
Hog hair, between sheets of paper	5.76	71	0.26	
Mineral wool	10	90	0.27	
Sawdust	12	90	0.41	
Shavings	8.8	90	0.41	
Sugar cane, fiber board	13.2	90	0.34	
Wood, balsa, across grain	7.3	90	0.41	
	8.8	90	0.38	
	20.0	90	0.58	
Wood fiber, board	16.9	90	0.34	
Vermiculite, expanded	6.2		0.32	

to moisture content, the effect of which is to increase heat transfer by conduction.

Conductivities and conductances for representative building and insulating materials are given by Tables 10 and 11.

75. Conductances of Air Spaces. The value of the conductance *a* of an air space is dependent upon the following conditions: the mean temperature of the confined air and the amount of its movement; the distance between the bounding surfaces and their nature as regards emissivity and reflectivity; and the position of the space, i.e., horizontal or vertical. In general there is little increase in the values of the conductances *a* for spaces wider than one inch for a given mean air temperature. The conductances for an air space of fixed width increase as the mean temperature of the air increases. Data[5] perti-

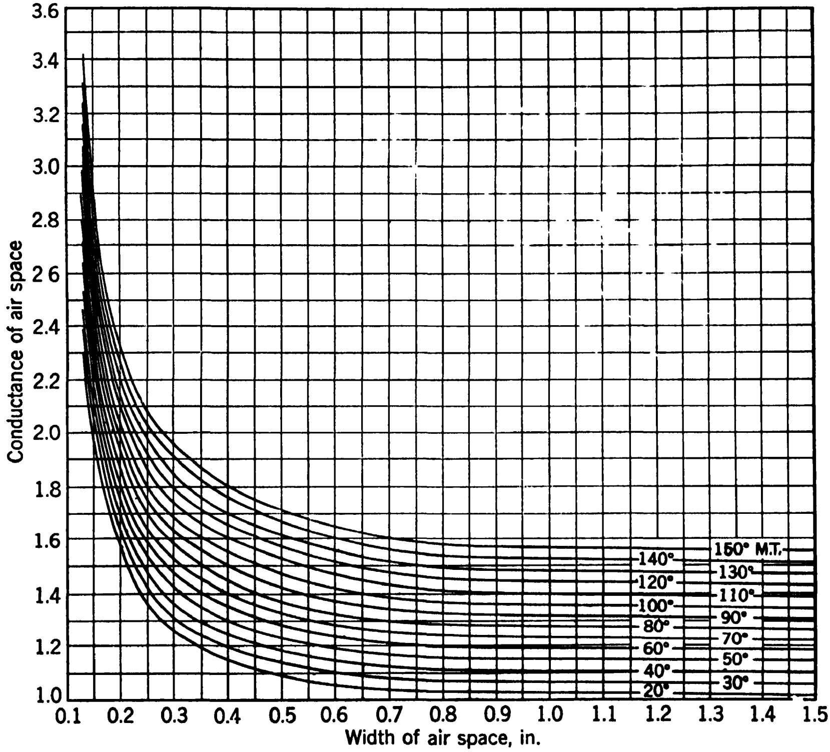

FIG. 31. Air-space conductances.

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[5] "Thermal Resistances of Air Spaces," by F. B. Rowley, A. B. Algren, and J. L. Blackshaw, *ASHVE Trans.*, Vol. 35, 1929.

nent to vertical air spaces, with non-reflective surfaces, are given by Fig. 31; also data for other air-space conditions are listed in Table 11.

If air spaces in building walls are to be effective in reducing heat-transmission losses they must be so constructed that the confined air

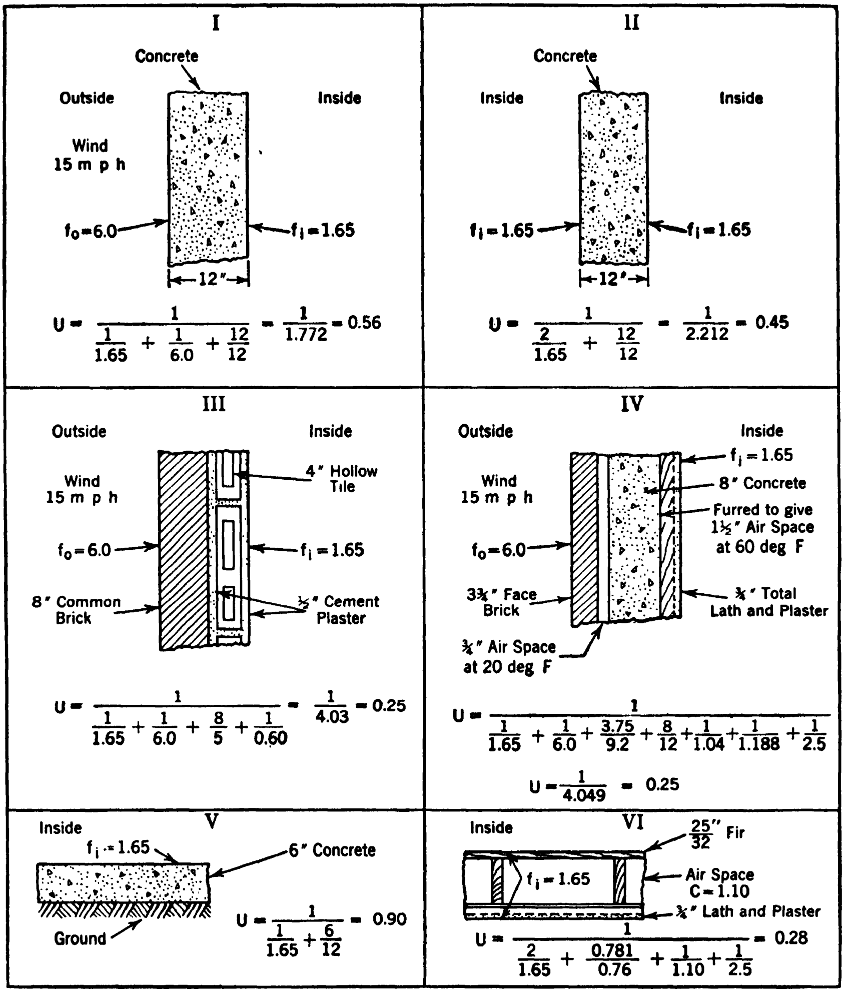


FIG. 32. Numerical calculations of *U*.

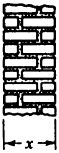
will have the minimum amount of motion. Air movement within wall spaces increases the transmission of heat by the process of convection.

76. Numerical Calculations of *U*. The examples in Fig. 32 may be taken as typical of the process of calculating overall coefficients of heat transmission *U*. In respect to the concrete wall, I and II with air

TABLE 12

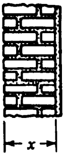
Coefficients of Heat Transmission, U, for Walls Embodying Masonry and Concrete Construction

Common brick



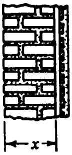
x	U
8"	0.42
12"	0.32
16"	0.25

Common brick with ½" plaster inside



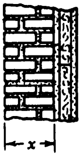
x	U
8"	0.40
12"	0.31
16"	0.25

Common brick, furred, ¾" air space, wood lath and plaster ¾" total



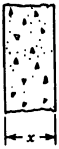
x	U
8"	0.27
12"	0.22
16"	0.19

Common brick, ½" cement mortar, 1½" corkboard, and ½" plaster



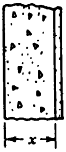
x	U
8"	0.12
12"	0.11
16"	0.10

Concrete



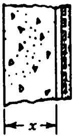
x	U
8"	0.70
12"	0.56
16"	0.48

Concrete with ½" plaster inside



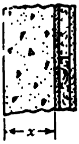
x	U
8"	0.65
12"	0.53
16"	0.44

Concrete, furred, ¾" air space, wood lath and plaster ¾" total



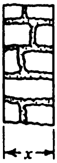
x	U
8"	0.36
12"	0.33
16"	0.29

Concrete, ½" cement mortar, 1½" corkboard, and ½" plaster



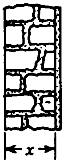
x	U
8"	0.14
12"	0.13
16"	0.12

Stone



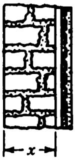
x	U
8"	0.71
12"	0.58
16"	0.49

Stone with ½" plaster inside



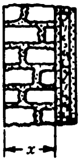
x	U
8"	0.66
12"	0.55
16"	0.46

Stone, furred, ¾" air space, wood lath and plaster ¾" total



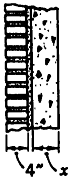
x	U
8"	0.37
12"	0.33
16"	0.30

Stone, ½" cement mortar, 1½" corkboard, and ½" plaster



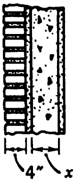
x	U
8"	0.14
12"	0.13
16"	0.13

4" face brick, ½" cement mortar, and concrete



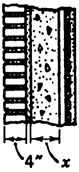
x	U
8"	0.52
12"	0.44
16"	0.38

4" face brick, ½" cement mortar, concrete, and ½" plaster



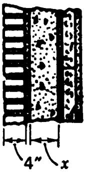
x	U
8"	0.49
12"	0.42
16"	0.37

4" face brick, ½" cement mortar, concrete, furred, ¾" air space, wood lath and plaster ¾" total



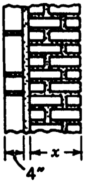
x	U
8"	0.31
12"	0.28
16"	0.26

4" face brick, ½" cement mortar, concrete, ½" cement mortar, 1½" corkboard, and ½" plaster



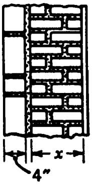
x	U
8"	0.13
12"	0.12
16"	0.12

4" cut stone, ½" cement mortar, and brick



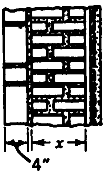
x	U
8"	0.36
12"	0.28
16"	0.23

4" stone, ½" cement mortar, brick, and ½" plaster



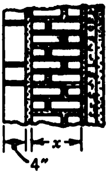
x	U
8"	0.35
12"	0.27
16"	0.22

4" stone, ½" cement mortar, brick, furred, ¾" air space, wood lath and plaster ¾" total



x	U
8"	0.25
12"	0.21
16"	0.18

4" stone, ½" cement mortar, brick, ½" cement mortar, 1½" corkboard, and ½" plaster

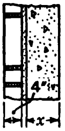


x	U
8"	0.12
12"	0.11
16"	0.10

TABLE 13

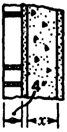
Coefficients of Heat Transmission, *U*, for Walls, Etc., Having Air Spaces

4″ cut stone, ½″ cement mortar, and concrete



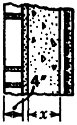
x	*U*
8″	0.55
12″	0.46
16″	0.40

4″ cut stone, ½″ cement mortar, concrete, and ½″ plaster



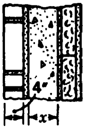
x	*U*
8″	0.51
12″	0.43
16″	0.38

4″ stone, ½″ cement mortar, concrete, furred, ¾″ air space, wood lath and plaster ¾″ total



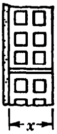
x	*U*
8″	0.32
12″	0.29
16″	0.26

4″ stone, ½″ cement mortar, concrete, ½″ cement mortar, 1½″ corkboard, & ½″ plaster



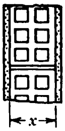
x	*U*
8″	0.13
12″	0.12
16″	0.12

¾″ stucco & hollow tile



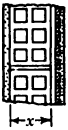
x	*U*
8″	0.40
12″	0.30
16″	0.25

¾″ stucco, hollow tile, and ½″ plaster



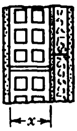
x	*U*
8″	0.38
12″	0.29
16″	0.24

¾″ stucco, hollow tile, furred, ¾″ air space, wood lath & plaster ¾″ total



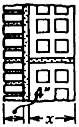
x	*U*
8″	0.26
12″	0.22
16″	0.19

¾″ stucco, hollow tile, ½″ cement mortar, 1½″ corkboard, and ½″ plaster



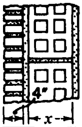
x	*U*
8″	0.12
12″	0.11
16″	0.10

4″ face brick, ½″ cement mortar, and hollow tile



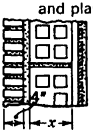
x	*U*
8″	0.34
12″	0.27
16″	0.22

4″ face brick, ½″ cement mortar, hollow tile, and ½″ plaster



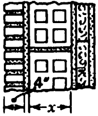
x	*U*
8″	0.32
12″	0.26
16″	0.21

4″ face brick, ½″ cement mortar, hollow tile, furred, ¾″ air space, wood lath and plaster ¾″ total



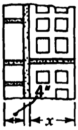
x	*U*
8″	0.24
12″	0.20
16″	0.17

4″ face brick, ½″ cement mortar, hollow tile, ½″ cement mortar, 1½″ corkboard, and ½″ plaster



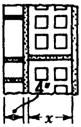
x	*U*
8″	0.11
12″	0.10
16″	0.09

4″ cut stone, ½″ cement mortar, and hollow tile



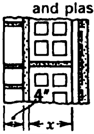
x	*U*
8″	0.36
12″	0.27
16″	0.23

4″ stone, ½″ cement mortar, hollow tile, and ½″ plaster



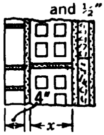
x	*U*
8″	0.34
12″	0.26
16″	0.22

4″ stone, ½″ cement mortar, hollow tile, furred, ¾″ air space, wood lath and plaster ¾″ total



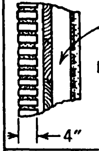
x	*U*
8″	0.24
12″	0.20
16″	0.18

4″ stone, ½″ cement mortar, hollow tile, ½″ cement mortar, 1½″ corkboard, and ½″ plaster



x	*U*
8″	0.12
12″	0.11
16″	0.10

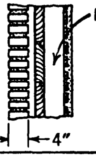
4″ face brick, ¾″ air space, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, wood lath and plaster ¾″ total



No insulation

U = 0.24

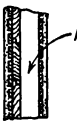
4″ face brick, ¾″ air space, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, wood lath and plaster ¾″ total



Insulation vermiculite 3⅝″ thick

U = 0.07

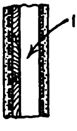
Stucco & wood lath ¾″ total, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, wood lath and plaster ¾″ total



No insulation

U = 0.34

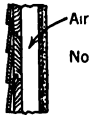
Stucco & wood lath ¾″ total, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, wood lath and plaster ¾″ total



Insulation vermiculite 3⅝″ thick

U = 0.08

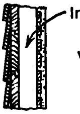
Clapboard or wood shingles ave. ⅝″ thick, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, lath and plaster ¾″ total



No insulation

U = 0.29

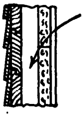
Clapboard or wood shingles ave. ⅝″ thick, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, lath and plaster ¾″ total



Insulation vermiculite 3⅝″ thick

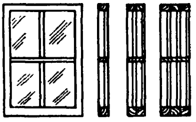
U = 0.08

Clapboard or wood shingles ave. ⅝″ thick, paper, 25/32″ Y.P. sheathing, 2″ x 4″ studding, 1½″ corkboard, and ½″ plaster



U = 0.12

Windows, skylights, and glass doors



Single *U* = 1.13
Double *U* = 0.45
Triple *U* = 0.28

TABLE 14

COEFFICIENTS OF HEAT TRANSMISSION *U* FOR WOODEN DOORS, INTERIOR WALLS, ROOFS, FLOORS, AND GLASS BLOCKS

Wooden Doors
Thin panel with or without glass 1.13.
Solid construction with thickness as stated below

Nominal Thickness	1 in.	$1\frac{1}{4}$ in.	$1\frac{1}{2}$ in.	$1\frac{3}{4}$ in.	2 in.	$2\frac{1}{2}$ in.	3 in.
Actual Thickness	$\frac{25}{32}$ in.	$1\frac{5}{16}$ in.	$1\frac{5}{16}$ in.	$1\frac{3}{8}$ in.	$1\frac{5}{8}$ in.	$2\frac{1}{8}$ in.	$2\frac{5}{8}$ in.
U	0.61	0.51	0.45	0.43	0.39	0.32	0.27

Flat Roofs without Ceilings

Construction	*U*		
	1 in. Nom. Th.	$1\frac{1}{2}$ in. Nom. Th.	2 in. Nom. Th.
Wood with ceiling joists and composition roofing	0.56	0.42	0.37
Wood, same as above with $\frac{7}{16}$-in. celotex insulation	0.32	0.27	0.25
Wood, same as above with 2-in. celotex insulation	0.13	0.12	0.12

Construction	*U*		
	4 in. Nom. Th.	5 in. Nom. Th.	6 in. Nom. Th.
Concrete with composition roofing	0.79	0.75	0.70
Concrete with 1-in. corkboard and composition roofing	0.35	0.34	0.33
Concrete with 2-in. corkboard and composition roofing	0.13	0.13	0.13

Wood Floors

Construction	*U*
$\frac{3}{4}$-in. plaster, wood lath, joists and $\frac{25}{32}$-in. YP flooring	0.31
Wood joists, $\frac{25}{32}$-in. YP subfloor, and $\frac{25}{32}$-in. oak or maple floor	0.37
$\frac{3}{4}$-in. plaster, wood lath, joists, $\frac{25}{32}$-in. YP subfloor and $\frac{25}{32}$-in. finish wood floor	0.25

Interior Walls

Construction	*U*
Studding lath and plaster	0.62
Plaster, lath, studding, lath and plaster	0.35
4-in. hollow tile, plastered one side	0.42
4-in. hollow tile, plastered both sides	0.40
4-in. hollow gypsum tile, plastered one side	0.28
4-in. hollow gypsum tile, plastered both sides	0.27

Pitched Roofs without Ceilings

Construction	*U*
Wood shingles on wood strips	0.50
Asphalt shingles on wood sheathing	0.59
Rigid asbestos shingles on wood sheathing	0.60
Slate or tile, roofing felt, and wood sheathing	0.61

Concrete Floor Slabs

Construction	*U*		
	4-in. Nom. Th.	6-in. Nom. Th.	8-in. Nom. Th.
Concrete on ground*	1.06	0.90	0.79
Concrete floor or ceiling slab, still air on both sides	0.65	0.58	0.53

Hollow-Glass-Block Walls

Construction	*U*, still air both sides	*U*, still air inside 15 mph outside
Smooth-surface glass blocks $7\frac{3}{4}$ in. × $7\frac{3}{4}$ in. × $3\frac{7}{8}$ in. thick	0.40	0.49
Ribbed-surface glass blocks $7\frac{3}{4}$ in. × $7\frac{3}{4}$ in. × $3\frac{7}{8}$ in. thick	0.38	0.46

YP = yellow pine

* A coefficient of 0.10 Btu is recommended for all types of concrete floor laid on the ground, with and without insulation, until more complete data are available.

motion at one surface, with still air at the other, and also with still-air conditions at both surfaces it will be noted that the effect of the air motion is to increase the overall coefficient of heat transmission U. The amount of the change is more pronounced, for any fixed air velocity, with thin walls than with thick walls.

Representative numerical data for U for different pieces of building construction are given by Tables 12, 13, and 14. The coefficients shown by both Fig. 32 and Tables 12, 13, and 14 are results obtained with $f_i = 1.65$, $f_o = 6$, and values of k and C as given in Tables 10 and 11. Thermal conductivities used are: average concrete, 12; cement mortar, 8; corkboard, 0.26; and plaster, 5.0. The conductance for an air space is 1.10 and for $\frac{3}{8}$-in. wood lath plus $\frac{3}{8}$-in. plaster, 2.5.

77. Combination Coefficients. When an attic space is unheated a combined coefficient of heat transmission for the roof above and the ceiling below it may be estimated. This combined coefficient is used with the ceiling area and the difference between the temperature of the air below the ceiling and that of the outside air, when calculations of the loss of heat through an attic space are made. The combined coefficient, *per square foot of ceiling area*, for a ceiling, an attic space, and a roof is

$$U_{cr} = \frac{U_r \times U_c}{U_r + \dfrac{U_c}{r}} \tag{50}$$

where U_r = coefficient of heat transmission for the roof.
U_c = coefficient of heat transmission for the ceiling.
r = ratio of roof area to ceiling area.

When still-air conditions prevail in the attic the radiation of heat through the air of the space is compensated by increasing the surface conductance of the roof and ceiling areas within the space from 1.65 (for still air) to 2.20 when computing U_r and U_c. If the air is in motion within the space a surface conductance larger than 2.20 is required. When the attic air motion is rapid because of a large amount of air infiltration, the ceiling area, the ceiling coefficient of heat transmission, the air temperature beneath the ceiling, and the outdoor air temperature must be used in calculating the heat losses which occur at ceiling areas.

78. Air Infiltration. The leakage of air into and out of a building may be the result of the action of wind, of the differential in temperature between the inside and the outside air with a resultant difference in their densities, or of the chimney action of tall structures. These factors may operate separately or in combination. Outward leakage is also the result of maintaining the air under pressure in the structure as done in some systems of heating and ventilation. In any event, the air leaking out of the space is replaced by an equal weight of air which comes from the outside. Infiltration occurs as the result of air passing through porous walls, through cracks in the walls, and through cracks about window and door openings. Infiltration with the corresponding exfiltration produces an additional load on either the heating or the cooling plant of a building. In heating, the incoming air must be warmed to the temperature maintained within the room; the outgoing air carries with it a quantity of heat equal to that necessary to heat the inleaking air, and this heat is lost from the structure. In summer cooling, air entering a room as a result of infiltration must be cooled to room temperature and its moisture content reduced when necessary.

79. Equivalent Wind Velocities. For low buildings the chimney or stack effect on air infiltration is generally negligible, but for tall structures consideration should be given to it. Passageways, such as elevator shafts and stair wells, should be isolated from adjacent spaces in order to reduce the effects of chimney action. The general results from the upward flow of air in a tall building are to promote the inward flow of air at the lower-story levels and to give diminished leakage effects at the upper sections. The foregoing action comes from the combined results of air-temperature difference and stack height, which produce a motive head that adds to wind effects at low levels and subtracts from them at higher elevations. Data presented later, Tables 16 and 17, show the marked effect of wind velocities on air infiltration through walls and at windows. A scheme of making allowances for the stack effect of a building is to compute equivalent wind velocities for use in estimating the air infiltration at different levels.

The mid-height of a building is usually taken as the location of the neutral zone[6] where there is no air-pressure differential between the outside and the inside spaces. When the neutral zone is used as a datuum level, equivalent wind velocities can be estimated for use in determining crack and wall air leakages. Equations which may be used for finding equivalent wind velocities at various distances above and below the neutral zone are

$$v_e = \sqrt{v^2 - 1.75a} \tag{51}$$

$$v_e = \sqrt{v^2 + 1.75b} \tag{52}$$

---

[6] "Neutral Zone in Ventilation," by J. E. Emswiler, *ASHVE Trans.*, Vol. 32, 1926.

where v_e = equivalent wind velocity, mph.
v = wind velocity existent at the considered location, mph.
a = distance of location above mid-height of building, ft.
b = distance of location below mid-height of building, ft.

80. Number of Air Changes per Hour Due to Infiltration. Commonly quoted values for the number of air changes per hour in an enclosed space due to natural causes are given in Table 15 for a room or building with various exposures and also for various classes of service.

TABLE 15

NUMBER OF AIR CHANGES PER HOUR EXCLUSIVE OF VENTILATION AIR SUPPLY*

Room or Building	Number of Air Changes per Hour n in Terms of Room Volume	Room or Building	Number of Air Changes per Hour n in Terms of Room Volume
One side exposed	1	Hall, entrance	2 to 3
		reception	2
Two sides exposed	$1\frac{1}{2}$	Living and dining rooms	1 to 2
Three sides exposed	2	Stores	1 to 3
Four sides exposed	2	Churches and factories	$\frac{1}{2}$ to 3
Without windows or outside doors	$\frac{1}{2}$ to $\frac{3}{4}$	Bathrooms	2

* *ASHVE Guide*, 1937 and 1948.

Never less than $\frac{1}{2}$ air change per hour should be considered.

81. Air Infiltration by Measurement. A number of investigations[7] have been made in laboratories and in actual buildings at various places in an endeavor to ascertain the probable leakage of air through the cracks about both steel and wood window sashes when subjected to different wind velocities. Laboratory investigations have also been carried on to determine the leakage of air through various wall sections when under air pressures equivalent to those of wind action.

A representative type of laboratory apparatus used for air infiltration investigations is shown by Fig. 33. The essential features of the set-up are a blower to supply air under pressure in chamber *A*; the wall section, with or without a window, placed between chambers *A* and *B*; an orifice box *C* having a carefully calibrated orifice plate; the necessary air-pressure gages; and an air-temperature thermometer. The air-pressure differential maintained between chambers *A* and *B* is indicative of that produced as the result of wind velocity.

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[7] "Air Infiltration through Various Types of Brick Wall Construction," by G. L. Larson, D. W. Nelson, and C. Braatz, *ASHVE Trans.*, Vol. 35, 1929.

"Air Infiltration through Double-Hung Wood Windows," by G. L. Larson, D. W. Nelson, and R. W. Kubasta, *ASHVE Trans.*, Vol. 37, 1931.

"Air Leakage on Metal Windows in a Modern Office Building," by F. C. Houghten and O'Connell, *ASHVE Trans.*, Vol. 30, 1924.

"The Weathertightness of Rolled Steel Windows," by J. E. Emswiler and W. C. Randall, *ASHVE Trans.*, Vol. 34, 1928.

"Air Leakage about Window Openings," by C. C. Schrader, *ASHVE Trans.*, Vol. 30, 1924.

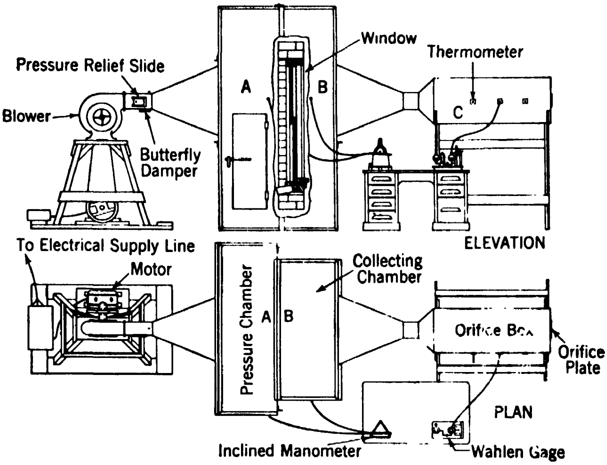


FIG. 33. Apparatus for testing window leakage.

TABLE 16

AIR INFILTRATION THROUGH WALLS*

Expressed in cubic feet per square foot per hour†

Type of Wall		Wind Velocity, MPH					
		5	10	15	20	25	30
8½-in. brick wall	Plain	2	4	8	12	19	23
	plastered‡	0.02	0.04	0.07	0.11	0.16	0.24
13 in. brick wall	plain	1	4	7	12	16	21
	plastered‡	0.01	0.01	0.03	0.04	0.07	0.10
	plastered§	0.03	0.10	0.21	0.36	0.53	0.72
Frame wall, with lath and plaster‖		0.03	0.07	0.13	0.18	0.23	0.26

[*] From "*Heating Ventilating Air Conditioning Guide 1948.*" Used by permission.

[†] The values in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed in chapter footnotes.

[‡] Two coats prepared gypsum plaster on brick.

[§] Furring, lath, and two coats prepared gypsum plaster on brick.

[‖] Wall construction: bevel siding painted or cedar shingles, sheathing, building paper, wood lath, and three coats gypsum plaster.

The leakage of air into a structure increases rapidly with an increase of wind velocity. A considerable quantity of air, exclusive of that entering about window and door openings, may pass through masonry and other types of walls which do not have some sealing material, such as plastering or its equivalent. The air leakage through brick walls which are unfinished on the inside may be a considerable amount but is reduced to a negligible figure by the addition of plastering to the inside faces as shown by Table 16.

Air leakage between window frames and masonry walls can be diminished by calking the cracks with plastic materials. The leakage about window frames and their sashes is dependent upon the wind velocity, the kind and type of wood or steel sash, and the crack width together with the sash clearance where the crack is. Sash clearance $F-E$ and sash crack C for movable wooden sash are defined by Fig. 34. A survey of a considerable number of buildings indicates that with double-hung wood sashes the following widths of crack and clearance for windows not weather-stripped may be considered as representative: (1) average window $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance; (2) poorly fitted windows $\frac{3}{32}$-in. crack and $\frac{3}{32}$-in. clearance.

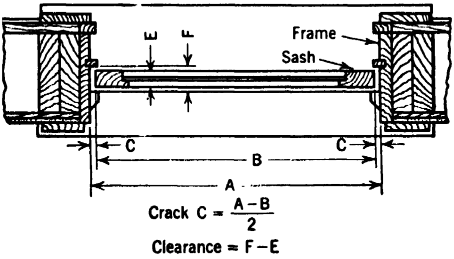


Fig. 34. Window crack and clearance.

Numerical quantities are given in Table 17 for the infiltration of air through the cracks about various kinds of windows. The volumes in cubic feet as given are per linear foot of window crackage, and although their conditions as to pressure and temperature are not specified the outside-air conditions may be used as being applicable to them. Attention is called to the reduction of air leakages by the use of weather stripping which is discussed in a following article.

The leakage of air at doors varies with their fit and warpage. For heating-load estimations, well-fitted doors may have about the same leakage per foot of crack considered, as poorly fitted double-hung sash and poorly fitted doors should have an allowance of twice the amount used for well-fitted doors. The foregoing values may be reduced by one-half if weather stripping is used. An outside single door, which is used frequently, may be considered to have three times the leakage of a well-fitted one.

TABLE 17

AIR INFILTRATION THROUGH WINDOWS*

Expressed in cubic feet per foot of crack per hour†

Type of Window	Remarks	Wind Velocity, MPH					
		5	10	15	20	25	30
Double-hung (wood sash windows (unlocked)	Around frame in masonry wall—not calked‡	3	8	14	20	27	35
	Around frame in masonry wall—calked‡	1	2	3	4	5	6
	Around frame in wood frame construction‡	2	6	11	17	23	30
	Total for average window, non-weather-stripped, $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance.§ Includes wood frame leakage‖	7	21	39	59	80	104
	Ditto, weatherstripped‖	4	13	24	36	49	63
	Total for poorly fitted window, non-weather-stripped, $\frac{3}{32}$-in. crack and $\frac{3}{32}$-in. clearance.¶ Includes wood frame leakage‖	27	69	111	154	199	249
	Ditto, weatherstripped‖	6	19	34	51	71	92
Double-hung metal windows**	Non-weather stripped, locked	20	45	70	96	125	154
	Non-weather stripped, unlocked	20	47	74	104	137	170
	Weather stripped, unlocked	6	19	32	46	60	76
Rolled section steel sash windows¶¶	Industrial pivoted, $\frac{1}{16}$-in. crack††	52	108	176	244	304	372
	Architectural projected, $\frac{1}{32}$-in. crack‡‡	15	36	62	86	112	139
	Architectural projected, $\frac{3}{64}$-in. crack‡‡	20	52	88	116	152	182
	Residential casement, $\frac{1}{64}$-in. crack§§	6	18	33	47	60	74
	Residential casement, $\frac{1}{32}$-in. crack§§	14	32	52	76	100	128
	Heavy casement section, projected, $\frac{1}{64}$-in. crack‖‖	3	10	18	26	36	48
	Heavy casement section, projected $\frac{1}{32}$-in. crack‖‖	8	24	38	54	72	92
Hollow metal, vertically pivoted window**		30	88	145	186	221	242

* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

† The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed in chapter footnotes.

‡ The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests.

§ The fit of the average double-hung wood window was determined as $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance by measurements on approximately 600 windows under heating season conditions.

‖ The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

¶ A $\frac{3}{32}$-in. crack and clearance represent a poorly fitted window, much poorer than average.

** Windows tested in place in building.

†† Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

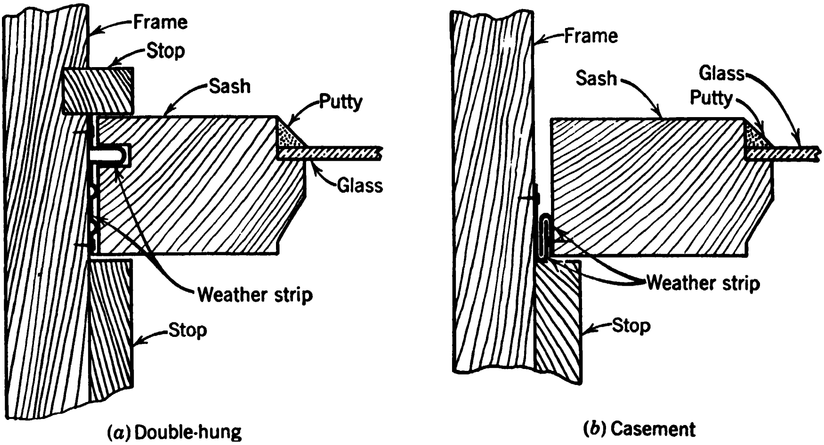
‡‡ Architecturally projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. $\frac{1}{32}$-in. crack is obtainable in the best practice of manufacture and installation, $\frac{3}{64}$-in. crack considered to represent average practice.

§§ Of same design and section shapes as so-called *heavy section casement* but of lighter weight. $\frac{1}{64}$-in. crack is obtainable in the best practice of manufacture and installation, $\frac{1}{32}$-in. crack considered to represent average practice.

‖‖ Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. $\frac{1}{64}$-in. crack is obtainable in the best practice of manufacture and installation, $\frac{1}{32}$-in. crack considered to represent average practice.

¶¶ With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With $\frac{3}{64}$-in. crack, representing poor installation, leakage at contact with steel framework is about one-third and at mullions about one-sixth of that given for industrial pivoted windows in the table.

82. Weather Stripping and Storm Sash. Air leakage or infiltration about window sash and doors may be materially reduced by the installation of weather stripping about their perimeters. Such building equipment may be of felt and rubberized strips installed to seal cracks, pieces of thin spring bronze attached to window sash and doors to bear against their enclosing frames, and some form of tongue-and-groove construction fabricated from strips of either sheet zinc, brass, or bronze. Two forms of weather stripping are shown by Fig. 35. The details of a weather-stripping installation depend upon the type of window served, that is, either double hung or casement, and the particular design of the weather stripping.



(*a*) Double-hung (*b*) Casement

FIG. 35. Installations of metal weather strips.

Data pertinent to the leakages of air at various wind velocities are given in Table 17 for three different cases of window construction operating without and with weather stripping. Percentages of reduction of air leakages occasioned by the use of weather stripping are given in tabular form in Table 18 for double-hung windows. This particular form of window construction is arranged so that the upper or lower sash may be raised or lowered independently of the other. A double-hung window has three horizontal and two vertical cracks through which air leakage may occur.

A study of the data of Table 18 indicates that the percentages of reduction of air leakage, resulting from the use of weather stripping, are variable and dependent upon the conditions of installation and operation. The heat losses from a building due to air infiltration change directly as a change in the amount of air infiltration occurs.

TABLE 18

PERCENTAGES OF REDUCTION OF AIR LEAKAGES RESULTING FROM THE USE OF WEATHER STRIPPING WITH DOUBLE-HUNG WINDOWS

Wind Velocity 15 mph

Type of Window	Condition	Leakage, Non-Weather Stripped, Cu Ft per Hr	Leakage Weather Stripped, Cu Ft per Hr	Per Cent Reduction
Double-hung wood sash, unlocked	Average	39	24	$\frac{(39-24)100}{39} = 38.4$
Double-hung wood sash, unlocked	Poor	111	34	$\frac{(111-34)100}{111} = 69.4$
Double-hung metal sash, unlocked		74	32	$\frac{(74-32)100}{74} = 56.7$

However, their percentage of the total heat losses per hour does not remain constant for all conditions.

Weather stripping applied to a window opening has no effect on the amount of heat lost per hour through its glass areas, but it does materially aid in keeping dirt carried by outside air from entering the structure and the loss of water vapor held by the inside air when humidification is necessary in winter air conditioning.

Storm windows fall under two general types; those which fill the window opening and which bear against the blind stops of the window frames and those which are clamped to the window sash and cover the glass areas. When this latter type of installation is made the storm sash does not reduce the infiltration of air at the window opening. In such cases the windows should be fitted with weather stripping. Storm windows fitted with felt gaskets and drawn tightly against the blind stops of window frames are of value in reducing air infiltration. However, if a storm sash is fitted loosely, no reduction of air infiltration may take place. For tight-fitting storm sashes the reduction of air infiltration is about the same as when weather stripping is used.

Storm sashes are quite effective in reducing the amount of heat transmitted per hour through window-glass areas. The coefficient of heat transmission U for two panes of glass separated by an air space is 0.45 Btu per hr per sq ft per deg F (Table 13), which is less than one-half that of a single thickness of glass. Under winter-weather conditions when storm sashes are used the inside-surface temperature of the glass is higher than that of a single pane of glass under the same conditions of temperature for inside and outside air. Two additional benefits accrue from this condition. The loss of radiant heat from the bodies of room occupants to the cool glass surface is lessened, and more comfortable sensations are produced with a given dry-bulb temperature in the occupied space. With higher inside-glass-surface temperature the dew-point temperature of the water vapor of the inside air is not obtained so easily. This item allows a higher relative humidity to be maintained in the inside air without excessive condensation on the glass and thereby tends to promote more comfortable conditions at a given dry-bulb temperature.

Double glazing of window sashes is also used to reduce heat losses and gains through their glass areas. Panes of small size are available for insertion into sashes. These units employ two panes of glass placed a small distance apart and carefully sealed about their edges by metallic or other materials. The sealing is done in the process of manufacture and before the units are used. The air of the space is partially evacuated to reduce its movement between the two panes. Complete sealing about the edges of the glass surfaces is essential as air leakages permit dirt and water vapor to enter the space between the two panes of glass. Dirt deposits are likely to accumulate on the enclosed surfaces when the leakage of air to and from the separating space occurs. Such deposits cannot be removed from interior-surface areas. Likewise in cold weather, moisture produced by water-vapor condensation may collect on the interior surfaces to cloud them.

The heat transmission of a double-glazed window sash is about the same as that for a single-glass one fitted with a storm sash. The coefficient U may be taken as given in Table 13 for double-glass construction. Sashes fitted with double glazing have air leakages about their perimeters. Weather stripping should be used with double-glazed sashes to reduce the air leakages about their perimeters and to reduce either the heat losses or gains in the spaces served.

83. Linear Feet of Crack Used in Computing Air Leakage. The amount of air infiltration is dependent upon the wind velocity, the width of the cracks, the linear feet of cracks, etc., through which air inleakage occurs. Generally it is not possible to fix accurate values definitely for any or all of the foregoing items. Therefore, the calculation of air infiltration by any method must be at best a reasonable approximation. The number of linear feet of window and door cracks to be used may be empirically fixed as: rooms with one exposure, all the linear feet of cracks in the outside wall; rooms with two exposures, the linear feet of crack in the outside wall having the greater amount of cracks; and rooms with three or four exposures, the length of the cracks in the wall having the greatest amount; but in no case must less than one-half the total cracks in the outside walls be used. The foregoing rule is based on the assumption that the angle of incidence of the wind on the exposed side or sides may vary from 0 to 90 deg.

84. Heat Losses Due to Air Infiltration. Unless building walls are of poor construction air leakage through them, Table 16, may usually be ignored. The amount of air, Q cu ft per hr, leaking into a room is the product of the space volume, in cu ft, and its number, n, of air changes per hour. The accuracy of such calculations is affected by the quantity n which is often difficult to fix definitely.

The data of Tables 16 and 17 and the assumptions of Art. 83 provide data and information which lead to more accurate estimations of the amounts of air inleakage under existent conditions. The quantity Q is the product of L ft of effective cracks (Art. 83). and q_i the leakage of air in cubic feet of air per foot of crack per hour, Table 17, for a crack of a given width at a definite wind velocity in miles per hour. No matter how the quantity of inleaking air Q is estimated

$$H_i = c_{pa}Qd_a(t - t_o) \tag{53}$$

where H_i = infiltration loss, Btu per hr.
c_{pa} = mean specific heat of air, 0.24.
Q = volume of air, cu ft per hr.
d_a = density of air, lb per cu ft at t_o F. Often used as 0.075 for all conditions.
t = inside-air temperature, deg. F
t_o = outside-air temperature, deg F.

Based on crack leakage

$$H_i = c_{pa}q_id_aL(t - t_o) \tag{54}$$

As a matter of convenience, for a crack of a given width subjected to a definite wind velocity, the portion $c_{pa}q_id_a$ of equation 54 may be computed as a constant C_i whereby

$$H_i = C_iL(t - t_o) \tag{55}$$

85. Thermal Storage of Building Materials. Walls and roofs have the ability to store variable amounts of heat. This property is dependent upon the method of construction, the kinds of materials used, and the thickness of the individual component parts. The item of thermal storage is of importance in problems both of winter heating and of summer cooling.

Obviously, thick walls of a given material will store more heat than thin walls of the same material. During the winter the heat stored in heavy walls will tend to maintain the desired inside-air temperature during periods when the heating plant is not actively operating. On the other hand, when the building is intermittently heated, heavy thick walls which have become cold may maintain a low inside-surface temperature for a considerable period after the inside-air temperature has reached that value deemed necessary for comfort. Under such conditions the occupants of the structure may not feel comfortable until the walls have absorbed enough heat to bring their inside-surface temperatures to a more nearly normal condition and thereby lessen the loss of heat by radiation from the occupants to the inside wall surfaces. Where intermittent heating during cold-weather periods is carried on as in buildings operating with reduced air temperatures during the night or in buildings heated only once or twice per week the heating-up process should be started well in advance of the period of occupancy.

The effect of heat storage in walls, ceilings, and roofs during periods of hot weather is to delay the passage of heat into the interior spaces of the building. However, once the materials have become heated, the effect is to prolong the load on the air-conditioning plant after the outside-air temperatures have begun to decrease. Owing to this storage action the maximum flow of heat from the outside to the inside air may not occur at the time of day when the maximum outside-air temperature prevails. The heat stored in the walls may not be dissipated until far into the night, and during prolonged periods of hot weather the walls may not cool to the temperature of the night air. As a consequence the storage of heat will be completed at an earlier time the following day, and the load on the air-conditioning plant, due to heat from the outside, will begin to build up at an earlier hour. Inside wall-surface temperatures fall slowly during the night in hot weather even when cool night air is mechanically circulated through the building.

86. Building Insulation. A survey of the thermal conductivities k and the conductances C, given in Table 11, for insulating materials used in building construction indicates that such substances are resistant to the transmission of heat by conduction as their conductivities are much less than those of other building materials. Certain metals in thin sheets or foil form, such as tin foil or aluminum foil, are also resistant to the transmission of radiant heat owing to the reflecting powers of their bright surfaces. These materials may lose some of their power to repel radiant heat when their surfaces become tarnished or dirty. Many insulating materials owe their desirable properties to the numerous small air cells or air spaces enmeshed within them. In general, the less the density of an insulator, the lower is its value of heat conductivity; also important is its moisture content, as water held in the material tends to increase its thermal conductivity.

Basic materials from which building insulation is derived are asbestos, bagasse (crushed sugar-cane stalks), cork, cornstalks, cotton, eel-grass, gypsum, glass, hair, jute, kapok, limestone, metals (aluminum and tin foil, also thin steel sheets separated by air spaces), moss, mica, paper pulp, rubber, straw, and wood fiber in several forms.

Insulation may be divided into the following types: (1) rigid, (2) semi-rigid, (3) flexible, and (4) fill. *Rigid* insulation is made in panels of various sizes and thicknesses, and some of it has considerable structural strength. *Semi-rigid* insulations embrace those of the order of felts which have some flexibility and are available in panels of different widths, lengths, and thicknesses. *Flexible* insulation embodies the quilts and blankets having paper or fabric on each side with loosely packed material between the coverings. Included in this category are the metal foils which are used either in crumpled form as filling material or as thin sheets suspended in air spaces and spaced a slight distance apart. *Fill* insulation is produced in shredded, powdered, or granulated form and is used to fill spaces in walls, floors, and ceilings.

The use of sufficient amounts of an insulating material in building construction is justified from the standpoints of economy and comfort, and sometimes its use is a necessity. Part of the expenditure for insulation can be saved in the reduced first cost of either a heating or a cooling plant. A plant of smaller capacity is required in either case, if the heat losses or heat gains can be reduced by the use of building insulation, and the operating costs are less. Furthermore, the reduced transmission of heat through walls, ceilings, and roofs leads to higher inside-surface temperatures in the winter and lower inside-surface temperatures during the summer, both of which are an aid in promoting human comfort and at the same time lead to economy of plant operation. Certain industrial processes must be carried on in rooms having conditioned air of a high relative humidity. By increasing the building inside-surface temperatures through the use of building insulation the dew-point temperature of the air at the inside surfaces can be avoided and moisture condensation on the surfaces of walls, ceilings, and floors prevented.

The selection of building insulating materials cannot be justified on the basis of a low coefficient of heat conductivity alone. Certain materials having low coefficients of conductivity are more expensive per inch of thickness than others having a higher heat conductivity. Consequently, the latter may be used in a greater thickness to produce a wall having an equal or lower overall coefficient of heat transmission, U, at a smaller first cost than would be possible with the more expensive material of lower conductivity. Along with the first cost of the insulating material consideration should be given to its durability, its ease and cost of placement in the building, its hygroscopic properties, and the additional structural loads involved by its use.

Each wall, roof, floor, or ceiling is an individual problem in which an attempt should be made to reduce economically the overall coefficient of heat transmission to a low value. There is an economic thickness in each case beyond which the cost of additional material cannot be justified. The actual placement of the insulating material in the walls has much to do with the final results obtained. Where insulators of the rigid, semi-rigid, and flexible types are used the best results are obtained when the insulation is so placed that the number of air spaces in the walls is increased. By placing the insulation so that two or more insulation surfaces are exposed to the air, Fig. 36, additional resistances to heat flow can be obtained in addition to that secured by the thickness of the material or materials.

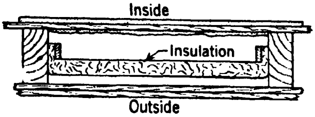


FIG. 36. Air space divided by insulation.

87. Vapor Barriers. Moisture in the air of heated spaces may accrue from various sources, such as cooking in kitchens, laundries, steam tables in restaurants, manufacturing operations, and that supplied by modern air-conditioning systems used to produce atmospheric comfort conditions. As the humidity ratio of an air-vapor mixture is increased its vapor pressure becomes greater. When the air outside of a heated space is cold its water vapor content may be low with a correspondingly low vapor pressure. Such conditions produce a vapor-pressure differential which tends to cause the water vapor of the inside air to flow outward through the cracks and porous materials of walls, ceilings, and floors. Figure 37 indicates permissible relative humidities[8] for various transmission coefficients.

Figure 29 indicates temperature gradients which may exist in walls of frame construction. The graphs show that the temperatures at various locations within the wall construction are materially below that of the warm room air. Whenever vapor passing from the interior of a space is cooled to its saturation temperature (dew-point temperature of air), condensation occurs. The point at which the formation of water takes place is dependent upon the wall construction. When the wall has a poor thermal resistance and the outside-air temperature is low, moisture may deposit on its inside surface exposed directly to the room air. Vapor penetration into walls of better construction may lead to the formation not only of moisture but also of frost and ice in their interiors. The walls of Fig. 29 show conditions favorable for the formation of ice and frost at the outer air-space surface. Moisture collection in a wall leads to increased heat transmission, deterioration of its materials, discoloration of finished surfaces, and the blistering and peeling of paint on areas so treated.

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[8] "Permissible Relative Humidities in Humidified Buildings," by Paul D. Close, *ASHVE Trans.*, Vol. 45, 1939.

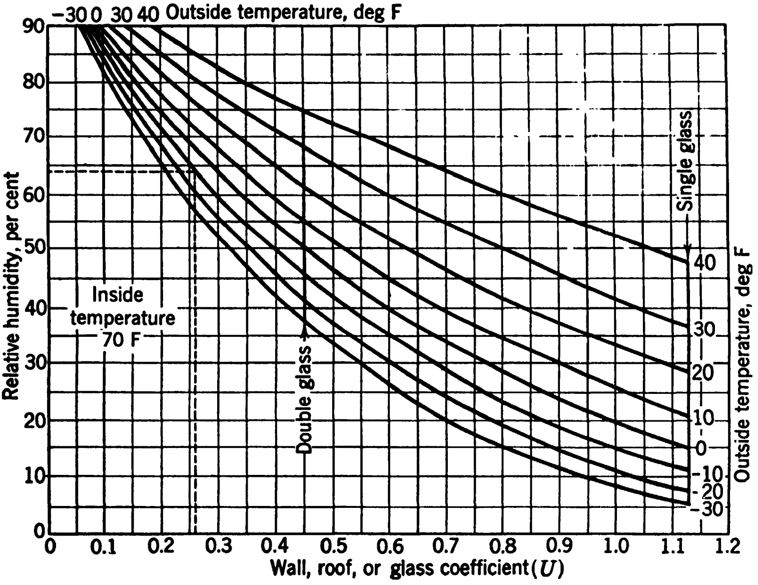

FIG. 37. Permissible relative humidities for various transmission coefficients.

Insulation placed within a wall, and other places, leads to reduced heat losses through it and a higher temperature of the surface exposed to the room air. Insulation also creates greater temperature variations from point to point in the path of heat flow so that vapor penetration into the interior of the wall section is highly undesirable. Where moisture penetration into a wall is likely to prove objectionable, provision should be made to prevent its occurrence. The amount of water vapor entering a wall and other sections, should not exceed 1.25 gr per hr, per sq ft, per in. of mercury vapor-pressure differential."[8] Vapor condensation in a space may be prevented by ventilation in attics and "by the use of some sort of a vapor barrier" in walls, ceilings, and floors.

Existing walls can be sealed against vapor penetration by filling cracks and holes and coating the visible inside surfaces with a compound of aluminum-bronze pigment and waterproof varnish. Waterproofed papers, impregnated with asphalt, provide resistance to vapor penetration in new walls. This material when used as a barrier should be located just behind the plastered interior as near as possible to the warm air of the room. Board-type plaster bases with the back of the material waterproofed are also available for new construction. Many insulating materials of the rigid, flexible, and bat types have incorporated with them either a coating or asphalt-treated paper on at least one of their surfaces to serve as a moisture barrier. All moisture or vapor barriers should have their joints sealed to prevent entrance of vapor into the wall interior through cracks at the edges of the barrier sheets.

88. Procedure in Making Heat-Loss Calculations. Certain data are necessary before the estimated losses of heat from a room or a structure may be calculated. Therefore, the following items must be either computed or fixed.

1. The separate net areas of the walls, glass, ceiling or roof, and floors through which the transmission of heat will occur.
2. The overall coefficients of heat transmission for the component parts of the building listed in item 1. These are to be taken from tables of data or calculated.
3. The inside-air temperature, at the breathing level 5 ft above the floor, which is considered to be necessary when severe winter weather exists; also the air temperatures at the proper levels for the calculation of heat losses through the various exposed building areas.
4. The outside-air temperature which is to be used for design purposes.
5. The computed losses of heat occurring at the various areas listed in item 1, based on the proper coefficients of heat transmission and the difference in temperature between the inside and the outside air.
6. An estimate of the heat required to warm the inleaking cold air based on the methods given in Art. 84.
7. The total of the individual transmission losses and those due to air leakage. This final summation gives the estimated heat losses from the space considered, for the air temperature conditions chosen, after the room or building has been heated.

89. Wind Movement during the Heating Season. In the discussion of the heat transmission of building materials and air infiltration, mention was made that the effect of air motion is to increase both. Consequently, the prevalent winds and their velocities are of considerable

TABLE 19

HEATING SEASON CLIMATIC DATA

Compiled from Records of the U. S. Weather Bureau and Other Sources

State	City	Average Temperature, Oct. 1–May 1	Lowest Temperature	Average Wind Velocity, Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.	Normal Degree Days, Total for Year
Ala.	Birmingham	53.9	−10	8.6	N	2618
	Mobile	57.7	− 1	8.3	N	1567
Ariz.	Flagstaff	34.9	−25	6.7	SW	7145
	Phoenix	59.5	16	3.9	E	1446
Ark.	Fort Smith	49.5	−15	8.0	E	3230
	Little Rock	51.6	−12	9.9	NW	3005
Calif.	Los Angeles	58.6	28	6.1	NE	1390
	San Francisco	54.3	27	7 5	N	3143
Colo.	Denver	39.3	−29	7.4	S	5863
	Grand Junction	39.2	−16	5.6	SE	5647
Conn.	New Haven	38.0	−15	9.3	N	5879
D. C.	Washington	43.2	−15	7.3	NW	4598
Fla.	Jacksonville	61.9	10	8.2	NE	1161
Ga.	Atlanta	51.4	− 8	11.8	NW	3002
	Savannah	58.4	8	8.3	NW	1647
Idaho	Lewiston	42.5	−13	4.7	E	4924
	Pocatello	36.4	−20	9.3	SE	6459
Ill.	Chicago	36.4	−23	17	SW	6287
	Springfield	39.9	−24	10.2	NW	5463
Ind.	Evansville	44.1	−15	8.4	S	4387
	Indianapolis	40.2	−25	11.8	S	5487
Iowa	Dubuque	33.9	−32	6.1	NW	6790
	Sioux City	32.1	−35	12.2	NW	6909
Kan.	Dodge City	40.2	−26	10.4	NW	5077
Ky.	Louisville	45.2	−20	9.3	SW	4428
La.	New Orleans	61.5	7	9.6	N	1208
	Shreveport	56.2	− 5	7.7	SE	2127
Me.	Eastport	31.1	−23	13.8	W	8451
	Portland	33.6	−17	10.1	NW	7338
Md.	Baltimore	43.6	− 7	7.2	NW	4522
Mass.	Boston	37.6	−18	11.7	W	5943
Mich.	Detroit	35.4	−24	13.1	SW	6580
	Marquette	27.6	−27	11.4	NW	8786
Minn.	Duluth	25.1	−41	11.1	SW	9766
	Minneapolis	29.6	−33	11.5	NW	7989
Miss.	Vicksburg	56.0	− 1	7.6	SE	2073
Mo.	Kansas City	42.8	−22	10.2	NW	4984
	St Louis	43.3	−22	11.8	NW	4610
	Springfield	43.0	−29	11.3	SE	4567
Mont.	Billings	34.7	−49	. . .	W	7119
	Havre	27.7	−57	8.7	SW	8474
Neb.	Lincoln	37.0	−29	10.9	N	6010
	North Platte	34.6	−35	9.0	W	6131
Nev.	Winnemucca	37.9	−28	9.5	NE	6371
N. H.	Concord	33.4	−35	6.0	NW	7391

TABLE 19 (*Continued*)

State	City	Average Temperature, Oct. 1–May 1	Lowest Temperature	Average Wind Velocity, Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.	Normal Degree Days, Total for Year
N. J.	Atlantic City	41.6	− 9	10.6	NW	5049
N. M.	Santa Fe	38.0	−13	7.3	NE	6124
N. Y.	Albany	35.1	−24	7.9	S	6658
	Buffalo	34.7	−21	17.7	W	6935
	New York	40.3	−14	13.3	NW	5306
N. C.	Raleigh	49.7	− 2	7.3	SW	3281
	Wilmington	53.1	5	8.9	SW	2432
N. D.	Bismarck	24.5	−45	...	NW	8969
Ohio	Cleveland	36.9	−18	14.5	SW	6171
	Columbus	39.9	−20	9.3	SW	5536
Okla.	Oklahoma City	48.0	−17	12.0	N	3698
Ore.	Baker	34.1	−20	6.0	SE	7219
	Portland	45.9	− 2	6.5	S	4379
Pa.	Philadelphia	41.9	−11	11.0	NW	4749
	Pittsburgh	40.8	−20	13.7	NW	5466
R. I.	Providence	37.6	−17	14.6	NW	6014
S. C.	Charleston	56.9	7	11.0	N	1870
	Columbia	53.7	− 2	8.0	NE	2504
S. D.	Huron	28.1	−43	11.5	NW	7997
	Rapid City	32.3	−34	7.5	W	7225
Tenn.	Knoxville	47.0	−16	6.5	SW	3665
	Memphis	50.9	− 9	9.6	NW	3078
Tex.	El Paso	53.0	− 2	10.5	NW	2538
	Fort Worth	54.7	− 8	11.0	NW	2356
	San Antonio	60.7	4	8.2	N	1424
Utah	Modena	38.1	−24	8.9	W	6605
	Salt Lake City	40.0	−20	4.9	SE	5637
Vt.	Burlington	29.3	−30	12.9	S	7930
Va.	Lynchburg	45.2	− 7	5.2	NW	4082
	Norfolk	49.1	− 2	9.0	N	3385
	Richmond	47.4	− 3	7.4	S	3944
Wash.	Seattle	45.3	3	9.1	SE	4864
	Spokane	37.5	−30	5.2	SW	6305
W. Va.	Elkins	38.8	−21	4.8	W	5814
	Parkersburg	41.9	−27	6.6	S	5091
Wis.	Green Bay	28.6	−36	12.8	SW	7956
	La Crosse	31.2	−43	5.6	NW	7442
	Milwaukee	33.0	−25	11.7	W	7086
Wyo.	Cheyenne	33.9	−38	13.3	NW	7549
	Lander	28.9	−36	3.0	NE	8237

importance in the estimation of the heat losses from a building in any locality.

Climatic data for various cities in the United States are given in Table 19. Rarely do the maximum wind velocity and the minimum outside-air temperature occur at the same time during the heating season. Heating engineers generally base their calculations for overall coefficients of heat transmission U and the air infiltration losses on the average wind velocity during the three months of the most severe winter weather. A commonly used average value for wind velocity during the coldest part of the heating season is 15 mph. Calculations made on this basis are safe, as the heating load of a structure during periods of extremely low outside-air temperatures, with practically no wind, may not be as large as when the outside air is warmer and the wind velocity greater. The latter condition is due to the larger inleakage of more moderately cold air caused by the outside-air motion.

90. Outside-Air Temperature. Outdoor-air temperatures range differently with the locality and the season of the year; they also may vary considerably over a period of hours in any location at any season. In the design of heating plants it is not customary to base the estimate of the heating requirements on the lowest outside-air temperature on record in any locality as this low temperature may prevail for only a few hours during the winter. Plants designed on the basis of a conservative outside-air temperature usually have enough reserve capacity together with the thermal storage in the building walls to carry through the periods of the lowest outside-air temperatures. Therefore, in order to avoid installing an excess of heating capacity, which would operate at only a fractional part of its rating during most of the heating season, the outside-air temperature for a heating-design basis is not taken as the lowest on record for the particular locality where the plant is to be installed.

The usual procedure is to secure United States Weather Bureau records of the lowest air temperatures and the average wind movement, together with its prevalent direction, during the heating season for the preceding ten years. For design purposes the outside-air temperature may be taken as 10 to 15 deg F above the lowest temperature on record for the location of the plant. Table 19 gives United States Weather Bureau data for various cities of the United States. If weather data are not available for the immediate location under consideration, statistics for the nearest city which has available records may be used.

91. Air Temperatures of Unheated Spaces. Design air temperatures of spaces such as attics, cellars, adjoining rooms, and spaces below floors, which are adjacent to heated rooms, are more difficult to fix than those of heated spaces. An empirical method of estimating the air temperature for use in the calculation of the loss of heat to an unheated space enclosure is to consider it as the mean of the air temperature of the heated space and that of the outside air. The air temperatures of unheated spaces vary with the heat-transmitting properties of the enclosing materials together with the air leakage of the enclosing construction. Numerical values of such temperatures range from 15 to 40 F.

The air temperatures, t_a, of adjacent unheated spaces may be estimated by means of equations expressing the flow of heat per hour, H_a, through the materials surrounding the space.

$$H_a = U_1A_1(t - t_a) = U_2A_2(t_a - t_o)$$

$$t_a = \frac{U_1A_1t + U_2A_2t_o}{U_1A_1 + U_2A_2} \tag{56}$$

where t = air temperature of the heated space, deg F.
t_a = air temperature of the unheated space, deg F.
t_o = outside-air temperature, deg F.
U_1 = overall coefficient of heat transmission of the floor, ceiling, or wall next to the warm space.
A_1 = area of the floor, ceiling, or wall bounding the heated space, sq ft.
U_2 = coefficient of heat transmission of the floor, ceiling, or wall exposed to outside air.
A_2 = area of the floor, ceiling, or wall exposed to the outside air, sq ft.

The above is true only when infiltration of air does not occur in the region under consideration. Allowances for the radiation of heat through the space are made by using larger conductances for the surfaces within the space than those considered satisfactory for still air conditions. If the air leakage into the unheated space is large in amount the air temperature is the same as that of the outdoor air.

For the estimation of building heat losses, the combination coefficients of heat transmission (Art. 77) and inside- and outside-air temperatures should be employed wherever possible.

92. Inside-Air Temperatures. The usual specified air temperatures in heating are either the breathing-line temperature, taken 5 ft above the floor, or the temperature at the 30-in. level and at a location not nearer than 3 ft from an outside wall in either case. The specified temperatures are dry-bulb values and vary with the purposes for which the heated space is used. Table 20 gives commonly specified breathing-line air temperatures for various classes of heating service.

Attention is called to Arts. 59 to 65 inclusive, where comfort conditions and the stratification of air are discussed. The designer must bear in mind the conditions of his design and select the proper inside-air temperature accordingly.

TABLE 20

BREATHING-LINE INSIDE-AIR TEMPERATURES FOR WINTER HEATING

Service	Dry-Bulb Temperature, Deg F	Service	Dry-Bulb Temperature, Deg F
Bathroom	85	Machine shop	60–65
Boiler shop	50–60	Operating room, hospital	85
Factory	65	Paint shop	80
Foundry	50–60	Public building	68–72
Garage	50–60	Residence	70
Gymnasium	55–60	School	70
Hospital	72–75	Store	65

93. Air Temperatures at the Proper Level. The estimation of the hourly flow of heat through areas of walls, windows, doors, floors, ceilings, and/or roofs should be based on the inside-air temperature at the proper level which is at the mean height of the surface under consideration.

The increase of air temperature above that of the breathing level is not directly proportional to the distance above a point 5 ft from the floor. The air temperature gradient curves of Figs. 93 and 95 show for a certain room the non-uniform rate of air temperature change with various changes of height of the points of observation. In view of the fact that authentic data are lacking for all conditions an approximation is necessary. The assumption is made that the increase of air temperature per foot of height above the 5-ft level is at the rate of 2 per cent of the breathing-line temperature up to ceiling heights of 20 ft. Beyond the level of 20 ft above the floor no further air temperature increases are to be considered as existent. In equation form the rule is expressed as

$$t' = t + 0.02(h - 5)t \tag{57}$$

where t' = air temperature at a level h feet above the floor, deg F.
t = breathing-line air temperature 5 ft above the floor, deg F.

The rule is satisfactory for rooms heated by tall direct radiators but does not hold so well for those systems of heating and ventilation which partially project heated air downward against its natural tendency to rise. Application of equation 57 should not be made in the calculations of rooms heated either by means of panels located in floors, walls, and ceilings or by use of radiant baseboard units. When the air temperature at the breathing line is not less than 55 F the air temperature at the floor may be taken as 5 F less than that 5 ft above the floor. When the room ceiling height is 10 ft or less corrections need to be made only for the floor and ceiling temperatures unless the window areas are placed unusually high in the walls.

Example. Find the air temperatures at the proper levels for the calculation of the heat losses of a room having a ceiling height of 18 ft, door openings 10 ft wide and 10 ft high, and window openings 4 ft wide and 8 ft high with the bottoms of the windows 3 ft above the floor. The breathing-line air temperature is to be 65 F.

Solution.

Air temperature at floor, $t_f' = 65 - 5 = 60$ F
Air temperature at ceiling, $t_c' = 65 + 0.02(18 - 5)65 = 81.9$ F
Air temperature at doors, $t_d' = 65 + 0.02(\frac{10}{2} - 5)65 = 65$ F
Air temperature at walls, $t_w' = 65 + 0.02(\frac{18}{2} - 5)65 = 70.2$ F
Air temperature at windows, $t_g' = 65 + 0.02[(3 + \frac{8}{2}) - 5]65 = 67.6$ F

When the roof serves as the ceiling of a room the air temperature under the roof is computed at the mean height of the roof, provided that the mean height does not exceed 20 ft. If the mean height exceeds 20 ft, the air temperature to be used under the roof is 1.3 t, where t is the breathing-line temperature.

94. Exposure Factors. During the heating season the wind velocities may be in excess of 15 mph and, although under such conditions the outside-air temperature is greater than that used in the design calculations, the heat losses due to transmission and infiltration may exceed those calculated for design purposes. Therefore engineers often add 15 per cent to the heat-transmission losses and infiltration losses occurring through the outside walls of rooms exposed to the prevailing winds. The use of an exposure factor is not necessary if the conditions of the design have been properly established.

95. Intermittent Heating. Economy of heating-plant operation is sometimes sought by reducing the output of the plant during the night or periods of time when the building is unoccupied. The room-air temperatures are not lowered to a point where water lines and plumbing fixtures may freeze, but the temperatures maintained are too low for human comfort.

Such a mode of operation will effect a saving in the cost of fuel, but it may also result in uncomfortable conditions in the structure when occupancy again takes place after a period of reduced air temperatures. The storage of heat in heavy building walls causes a lag in the drop of air temperature as the heating-plant output is reduced. Once the walls are cooled it may take a considerable period of time to warm them again to achieve comfort conditions with a given dry-bulb air temperature. When the air temperatures must be brought back to normal or even above normal to compensate for the effect of cold wall surfaces, in a short period of time, either the plant must be driven hard with reduced operating efficiency or else a plant of larger capacity than would be required with continuous heating must be installed.

The additional amount of heating capacity necessary is dependent upon the amount of heat stored in the walls, i.e., the wall thickness and its ability to hold heat and the length of time available for bringing the building up to comfort conditions. Walls which lose heat readily suffer large drops of their inner-surface temperatures during periods of reduced inside-air temperatures. When such conditions exist the wall surface temperature lag during reheating may make overheating of the air necessary to offset the loss of radiant heat from the bodies of the occupants to the cold wall surfaces. Therefore the savings theoretically made by intermittent heating may be largely wiped out. The additional allowances to be made in estimating the size of a heating plant operated intermittently depend upon local conditions and the heating engineer's judgment.

96. Numerical Example of the Calculation of Building Heat Losses. The procedure is as outlined in Art. 88. The current codes of the National Warm Air Heating and Air Conditioning Association and the Guide of the Institute of Boiler and Radiator Manufacturers give similar methods relative to the estimation of heat losses from structures.

When the building space is divided, the losses must be found for the individual rooms so that sufficient heat-disseminating capacity (radiators) may be provided for each separated space. The total heat losses from a building with several rooms is less than the sum of the estimated losses for the individual rooms. To find the heat losses from the building as a unit the transmission losses from all the rooms should be added together. This will give the heat losses exclusive of infiltration losses. The infiltration losses for the building as a unit should then be computed and added to the transmission losses. The total heat losses for the unit are to be used in the selection of the heating boiler or furnace and in the estimation of the fuel requirements for a season.

Example. Calculate for design purposes the heat losses from a room of a building as shown by Fig. 38.

Solution. Room volume = $24 \times 40 \times 13 = 12{,}480$ cu ft

Glass areas = $(9 \times 13.75) + (3.5 \times 9)5 = 281$ sq ft

Gross wall area = $13(24 + 40) = 832$ sq ft

Net wall area = $832 - 281 = 551$ sq ft

Floor or ceiling area = $24 \times 40 = 960$ sq ft

Window crack (worst side) = $5[(3 \times 3.5) + (2 \times 9)] = 142.5$ ft

$$U_w = \frac{1}{\dfrac{1}{1.65} + \dfrac{1}{6.0} + \dfrac{16}{5} + \dfrac{0.5}{5}} = \frac{1}{4.072} = 0.25$$

$$U_f = \frac{1}{\dfrac{1}{1.65} + \dfrac{6}{12}} = \frac{1}{1.106} = 0.90 \text{ (see footnote of Table 14).}$$

$$U_g = 1.13$$

$$U_c = \frac{1}{\dfrac{2}{1.65} + \dfrac{0.5}{5} + \dfrac{1}{0.26} + \dfrac{1}{1.10} + \dfrac{0.781}{1.03}} = \frac{1}{6.83} = 0.15$$

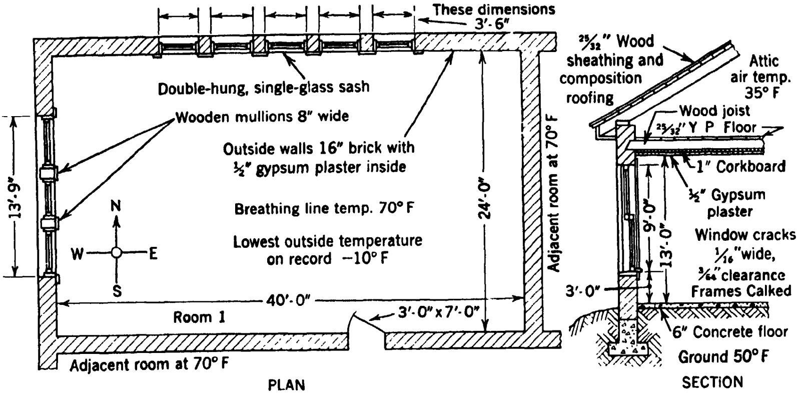

FIG. 38. Room for heat-loss calculations.

Infiltration of air per linear foot of crack, wind 15 mph = 39.3 cu ft at 0 F and 29.92 in. of mercury

Infiltration constant $C_i = 0.24 \times 39 \times 0.086 \times 1 = 0.81$ Btu

Outside-air temperature for design purposes $-10 + 10 = 0$ F

Breathing-line air temperature = 70 F

Air temperature at mean height of walls $= 70 + [0.02(\frac{13}{2} - 5)70] = 72.1$ F

Air temperature at mean height of windows $= 70 + [0.02(3 + \frac{9}{2} - 5)70] = 73.5$ F

Air temperature at ceiling $= 70 + [0.02(13 - 5)70] = 81.2$ F

$$\begin{aligned}
H_g &= 1.13 \times 281 \ (73.5 - 0) = 23{,}340 \\
H_w &= 0.25 \times 551 \ (72.1 - 0) = 9{,}930 \\
H_f &= 0.90 \times 960 \ (65 - 50) = 12{,}960 \\
H_c &= 0.15 \times 960 \ (82.1 - 35) = 6{,}780 \\
H_i &= 0.81 \times 142.5(73.5 - 0) = \underline{8{,}480} \\
&\text{Total hourly losses} = 61{,}490 \text{ Btu}
\end{aligned}$$

Where more than one room has to be considered it is convenient to arrange the data relative to the various rooms in the form of Table 21.

TABLE 21

HEAT LOSSES FROM A ROOM

Room Number	Air Temperature at Breathing Line, Deg F	Room Volume, Cu Ft	Glass Area, Sq Ft	Net Wall Area, Sq Ft	Floor Area, Sq Ft	Ceiling Area, Sq Ft	Linear Feet of Cracks	Heat Losses through Glass, Btu per Hr	Heat Losses through Walls, Btu per Hr	Heat Losses Floor, Btu per Hr	Heat Losses Ceiling, Btu per Hr	Infiltration Losses, Btu per Hr	Total Heat Losses, Btu per Hr
1	70	12,480	281	551	960	960	142.5	23,340	9930	12,960	6,780	8480	61,490

In the preceding example the actual heat losses through the floor slab are probably less than the 12,960 Btu per hr which were calculated and shown in Table 21. Earth beneath a floor laid upon it serves as a heat storage reservoir, and its moisture content, temperature, and thermal conductivity vary considerably. Because of these factors the coefficient $U_f = 0.90$ Btu, when used with a temperature difference of 15 F as in the example shown, may not give the actual amount of heat lost through a floor slab; the same is true when a coefficient equal to 0.10 Btu (Table 14) is employed.

Investigations,[9] at the University of Illinois, have indicated a procedure similar to the following one for an uninsulated floor slab resting on earth. Heat may be expected to flow to outside air through a floor area which has a width of 3 ft along the room walls, which have an outside exposure. For use in making calculations with an uninsulated floor resting on earth a factor of 0.58 may be used with the outside-air-temperature difference. The calculation of the heat lost through the portion of the slab which is 3 ft in width and L ft in length is $H_{fo} = 0.58 \times L(t_f - t_o) = 0.58(40 + 24)(65 - 0) = 2415$ Btu per hr. The remaining floor area $A_{fi} = 37 \times 21 = 777$ sq ft, and its loss of heat $= 0.10 \times 777(65 - 50) = 1165$ Btu per hr, when $U_f = 0.10$. This method of calculation indicates that the floor losses H_f are $H_{fo} + H_{fi} = 2415 + 1165 = 3580$ Btu per hr. This quantity is materially lower than the 12,960 Btu per hr as given in Table 21. This calculation changes the estimated heat losses of the room from 61,490 to 52,110 Btu per hr. Until more definite data are available the estimation of the losses of heat through floor slabs resting on earth is governed largely by the calculator's experience and judgment.

97. Heat Sources. The heat losses from a room or a building must be estimated to cover the design conditions of air temperatures, wind velocities, etc., without thought of the sources of heat within the space to be warmed. This routine is necessary in order to properly select a heating plant for the service to be rendered as there may be no gains of heat within a space during periods of unoccupancy. Therefore, the plant must be capable of producing a comfortable air condition within a heated space prior to its occupancy. After occupancy does take place as in crowded auditoriums and theaters the sensible heat from people, lights, motors, and other equipment may change the problem from that of supplying the building heat losses to that of providing an adequate amount of conditioned ventilating air for the removal of excess heat. The problem may thus become one of temperature regulation.

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[9] "Temperature and Heat Loss Characteristics of Concrete Floors Laid on the Ground," by H. D. Bareither, A. N. Fleming, and B. E. Alberty. A report of an investigation conducted by the Small Homes Council and the Department of Mechanical Engineering of the University of Illinois, August, 1948.

The heat from lights, motors, etc., is the product of the wattage input and 3.412 Btu, the heat equivalent of a watt-hour. An estimate of the sensible heat liberated by people can be based on the data of Art. 61. This item is the product of the heat given off per individual and the number of room occupants.

98. Heat for Air Humidification. The additional load imposed upon a heating plant due to air humidification can be estimated by the methods discussed in Arts. 49 and 50. This humidification load should include that necessary to humidify air leaking in by infiltration and that also necessary for a supply of conditioned ventilating air.

99. Monthly and Seasonal Loads. When either steam, gas, or electricity is purchased for heating purposes, the rate per unit used is often dependent upon the amount consumed per month, and the total costs per heating season are the sum of the monthly costs. With coal and oil purchases by contract at a uniform price for the entire season it is not necessary to estimate the monthly costs in finding the seasonal costs. In any event the calculated costs, either per month or per season, must be based on the expected heat requirements of the building per unit of time considered. The purchased commodity, steam, coal, gas, oil, or electricity, must provide for the heat losses of the building as a unit and other losses such as from piping. In buying fuel the efficiency of combustion has to be taken into account. Two methods are available for the estimation of heat or fuel consumption. The first method to be considered is based on the heat requirements of the building per unit of time involved as estimated for the average inside-outside-air-temperature difference for the period. The second method requires data relative to a term known as the degree day.

By the method first mentioned the heat requirements per period considered are

$$M = \frac{H_b(t_r - t_a)N}{t - t_o} \tag{58}$$

where M = heat losses, Btu per month or per season as required.
H_b = heat losses to be supplied in building, including piping losses, based on design conditions, Btu per hr.
t_r = inside-air temperature, deg F.
t_a = average outside-air temperature during the period considered.
N = number of hours heating service rendered, per month or per season.
t = inside-air temperature used for design purposes, deg F.
t_o = outside-air temperature used for design purposes, deg F.

Purchased steam may be expected to yield approximately 1000 Btu per lb when used in a heating system. The weight W_s of steam per unit of time considered is $M \div 1000$ or

$$W_s = \frac{H_b(t_r - t_a)N}{1000(t - t_o)} \tag{59}$$

When fuel is burned to produce the heat, the required amount is

$$W_f = \frac{M}{F_{cu} \times E} \tag{60}$$

where W_f = units of fuel required per time interval considered, i.e., tons of coal, cu ft of gas, gal of oil, etc.
M = total heat losses for the period, Btu
F_{cu} = heating value of the unit of fuel used, i.e., Btu per ton, per cu ft, per gal, etc.
E = efficiency of fuel utilization expressed decimally.

The term *degree day* was originated by the American Gas Association. It is based on the idea that heat is not required in a building maintained at 70 F when the average outside-air temperature, represented by the mean of the maximum and minimum outside-air temperatures for the day, does not fall below 65 F. Each degree that the average outside-air temperature falls below 65 F represents a degree day. Thus if the average of the maximum and the minimum temperatures of a day is 60 F the number of degree days is 5. The degree days thus determined are totaled and reported per month and per heating season. These totals vary with different localities.

Use of the degree-day factor is made by multiplying the degree days involved for a given period of time by either the steam or fuel consumption, per degree day, of the building under consideration. Data, Table 19 for the degrees days of various localities during the heating season are compiled for use by several different agencies, among them the United States Weather Bureau and the ASHVE. The unit consumption of steam or fuel used per degree day is entirely dependent upon the building, the fuel used, and the efficiency of operation of the heating plant. These variable factors make comparisons of individual buildings with others of the same classification somewhat difficult for the following reasons.

TABLE 22

STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS*

(Heating season only)

		Steam Consumption Pounds per Degree Day—65 F Basis§		
Building Classification	Number of Buildings Listed	Per *M* Cu Ft of Heated Space	Per *M* Sq Ft of Radiator Surface†	Per *M* Btu per Hr of Heat Loss‡
Apartments	16	1.78	97.5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1.32	64.2	...
Printing	7	1.25	105.5	...
Clubs and lodges	10	0.96	77.0	...
Retail stores	18	0.90	80.6	0.268
Theatres	6	0.90	75.0	0.498
Loft and manufacturing	16	0.89	72.3	0.283
Banks	7	0.88	45.2	...
Auto sales and service	8	0.83	62.2	...
Churches	6	0.58	49.4	...
Department stores	14	0.57	60.7	0.238
Garages (storage)‖	6	0.42	72.3	...
Offices (total)	35	1.09	70.0	0.283
Offices (heating only)	35	0.975	65.4	0.256

* Includes steam for heating domestic water for heating seasons only.

† Equivalent steam radiator surface.

‡ Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).

§ The figures are a numerical, not a weighted, average for the several buildings in each class.

‖ Based on zero consumption at 55 F.

The kind, type, construction, and daily period of occupancy during the heating season of a building are of importance. All fuels vary in heating value per unit of measurement, i.e., per pound or per cubic foot. Likewise the efficiency of utilization of the heat from a given fuel is quite different in many cases. Therefore the unit consumption of steam or fuel is variable for buildings of the same general classification.

Comparative data for the steam consumption of different types of buildings per degree day are given by Table 22.[10]

PROBLEMS

1. A test of a wall section with a guarded hot box gave the following data: U equal to 0.28 Btu per hr per sq ft per deg F; inside-air temperature, 100 F; outside-air temperature, 80 F; and area of wall section, 9 sq ft. Find the electrical input to the box in watts per hour.

2. A wall section 3 ft by 3 ft had the heat equivalent of 6.6 watt-hours pass through it each hour when the air temperatures adjacent to its sides were 85 F and 75 F. Find the overall coefficient of heat transmission U.

3. The continuous input of electrical energy to a guarded hot box was 8.5 watts. The wall section under test had an area of 9 sq ft, and the air temperature difference between its two sides was 21 F. Find the overall coefficient of heat transmission U.

4. Find the overall coefficient of heat transmission U for a plain common-brick wall 20 in. thick when the wind velocity is 15 mph and the inside air is still.

5. Find the overall coefficient of heat transmission U for a 12-in. average-concrete wall with $\frac{1}{2}$ in. of stucco on the outside when the outside-wind velocity is 10 mph and the inside air is still.

6. Find the overall coefficient of heat transmission U for a wall consisting of 4 in. of face brick, $\frac{1}{2}$ in. of cement mortar, 8 in. of stone, and $\frac{3}{4}$ in. of gypsum plaster. The outside-air velocity is 15 mph, and the inside air is still.

7. A wall has an overall coefficient of heat transmission of 0.18 Btu per sq ft per deg F when the outside-wind velocity is 15 mph and the inside surface is exposed to still air. Find the inside- and outside-surface temperatures when the inside- and outside-air temperatures are 73 F and -15 F, respectively. What is the total resistance of the wall?

8. A wall with a total resistance of 8.3 has a stucco outer surface and a smooth-plaster inside finish. The outside-wind velocity is 25 mph, and the inside air is still. Find the overall coefficient of heat transmission U and the inside- and outside-surface temperatures when the inside air is 65 F and the outside-air temperature is -15 F.

9. A wall is constructed of 4 in. of face brick and 6 in. of hollow tile with $\frac{1}{2}$ in. of cement plaster on each side of the tile. Find the inside- and outside-air temperatures when the inside-surface temperature of the wall is 65 F and the difference of inside- and outside-air temperatures is 15 F when the outside-air velocity is 15 mph.

10. A wall consists of 8 in. of face brick, $\frac{1}{2}$-in. air space, $\frac{25}{32}$-in. yellow-pine boxing, and studding $3\frac{5}{8}$ in. wide, the spaces between the studding being filled with mineral wool; the inside finish is gypsum board $\frac{1}{2}$ in. thick. Find the overall coefficient of heat transmission U when the outside-wind velocity is 10 mph.

11. A wall consists of 4 in. of face brick, $\frac{1}{2}$ in. of cement mortar, 8-in. hollow clay tile, an air space $1\frac{5}{8}$-in. wide, and wood lath and plaster totaling $\frac{3}{4}$ in. thickness. Find the overall coefficient of heat transmission U when the wind velocity is 15 mph.

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[10] "Heat Requirements of Buildings," by J. H. Walker and G. H. Tuttle, *ASHVE Trans.*, Vol. 41, 1935.

12. A cinder-concrete roof 8 in. thick is covered with asphalt shingles. Find its overall coefficient U when the wind velocity is 15 mph.

13. A roof with yellow-pine decking $1\frac{5}{8}$ in. thick is covered with composition roofing. The air beneath the roof is at 95 F dry-bulb temperature and has a relative humidity of 70 per cent. What thickness of sugar-cane fiber board must be placed beneath the wood to prevent moisture condensation when the outside-air temperature is −50 F and the wind velocity is 15 mph. Allow 2 deg F as a safety factor.

14. A wall has an overall coefficient of heat transmission U equal to 0.23 Btu per sq ft per hr per deg F. What is the conductance of the wall when its outside surface is exposed to a wind velocity of 15 mph and its inside surface is in contact with still air?

15. A factory building has a volume of 100,000 cu ft in which one-half air change per hour takes place. The air temperature at the breathing line is 68 F. Find the hourly heat losses due to infiltration when the outside air is at −10 F.

16. Find the heat losses due to air infiltration in a bathroom which has a volume of 650 cu ft when the outside-air temperature is −25 and the inside-air temperature is 85 F. The space has a single exposure.

17. A room has double-hung windows each having 21 lin ft of $\frac{3}{32}$-in. crack with $\frac{3}{32}$-in. clearance. Find the heat lost, Btu per hour per foot of crack and per window, when the inside-air temperature is 75 F, the outside air temperature 10 F, and the wind velocity 10 mph.

18. A room has two exposures which have 60 and 115 lin ft of crack in unlocked double-hung metal windows. The inside-air temperature is 70 F, and the outside-air temperature is −15 F. Find the heat losses due to air infiltration when the wind velocity is 20 mph.

19. A roof has a coefficient of heat transmission of 0.50 Btu per hr per sq ft per deg F, and the ceiling beneath it has a coefficient of 0.45. The roof and ceiling areas are 900 and 600 sq ft, respectively. Find the air temperature in the attic space when the outside-air temperature is −10 F and the air temperature beneath the ceiling is 70 F.

20. The height of a room ceiling is 12 ft and the breathing-line air temperature is 72 F. Find the air temperature at the floor, the mean height of the wall, and at the ceiling.

21. A room has windows 3 ft by 5 ft with the bottoms placed 24 in. from the floor. Find the mean temperature of the air on the glass if the room breathing-line air temperature is 75 F and the room is heated by 32-in. tubular radiators. Find the air temperatures at the floor and the ceiling if the ceiling height is 10 ft.

22. Calculate the hourly heat losses for the room of Fig. 38 for design purposes when the inside-air temperature is 72 F at the breathing line and the air temperature of the adjacent rooms is 55 F. The interior walls are of 12-in. common brick plastered on each side with gypsum material $\frac{1}{2}$ in. thick, and the wood door is 2 in. thick. The lowest outside-air temperature on record is 0 F, and the outside wind velocity may be taken for design purposes as 15 mph.

23. A single-story building has common-brick walls 12 in. thick plastered with cement mortar $\frac{1}{2}$ in. thick on the inside. The building is 50 ft by 200 ft exposed on all sides and has no ceiling except the roof. The floor laid on the ground is 8-in. stone concrete. The roof has a ridge along the longitudinal axis of the building which is 30 ft. above the floor level. The roof is of $1\frac{5}{8}$-in. fir decking covered with composition roofing. The height of the side walls is 18 ft. Each long side has twenty windows 3 ft wide and 8 ft high placed with their bottoms 3 ft from the floor. The ends each have two windows of the size and placement of those of the side walls and also wooden doors $2\frac{1}{2}$ in. thick fitted into an opening 12 ft wide and 10 ft high. Each doorway has two horizontal cracks 12 ft long and three vertical cracks 10 ft. high, each $\frac{1}{16}$ in. wide. (Use doors as industrial pivoted steel sash for estimating air leakage.) The windows are double-hung wood sashes with $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance. The inside-air temperature desired at the breathing line is 55 F and the lowest outside-air temperature on record is 20 F, with an average wind velocity during the heating season of 15 mph. Consider the ground temperature as 50 F. Calculate for design purposes the hourly heat losses from the building.

24. A building has hourly heat losses amounting to 125,000 Btu based on an inside temperature of 70 F and an outside temperature for design purposes of -10 F. The average outside-air temperature during a month of 30 days was 35 F. Find the number of tons of coal required for the month if fuel having a heating value of 10,670 Btu per lb is used with an efficiency of 55 per cent. Assume 24 hr per day for the heating service, and allow 25 per cent extra for unavoidable losses in the system.

25. During the heating season a building having a volume of 80,000 cu ft required 1200 Btu per 1000 cu ft of volume per degree day for heating. The total degree days for a season amounted to 5400. Find the cost of heating the building with oil, which cost 9 cents per gal, that weighs 7 lb and has a heating value of 18,500 Btu per lb, if the oil is utilized with an efficiency of 70 per cent.

CHAPTER 5

FUELS, COMBUSTION, DRAFT, CHIMNEYS, AND AUTOMATIC-BURNING EQUIPMENT

100. Fuels. Heat energy may be secured by the rapid oxidation of a fuel existing in either a natural or a prepared form and which may be in either a solid, a liquid, or a gaseous state.

The principal fuels used for heating are: coal of various kinds, sizes, and grades and its derivative coke; distillates of petroleum oils, known as fuel oils; and either natural or manufactured gas. Wood although used somewhat in various localities will not be considered. The chief heat-producing elements of any commercial fuel are carbon and hydrogen or the compounds of the two.

101. Coal. In its various states coal is the remains of vegetable materials which have been subjected to great heat and pressure. The composition of coals vary greatly according to the changes that they have undergone and the impurities which have entered into them.

Accurate analyses of coal may be either proximate or ultimate. The *proximate analysis* separates a coal into four parts, which are, by percentages, moisture, volatile matter, fixed carbon, and ash. Generally a separate determination of the sulfur is made and reported with the proximate analysis together with the heating or calorific value of the coal. An *ultimate analysis* gives the percentages of all the coal constituents, which are: carbon, hydrogen, nitrogen, oxygen, sulfur, moisture, and ash. Both forms of analysis are available, but the ultimate analysis is of more value when exhaustive studies of combustion are to be made. Either analysis may be reported in three ways: (1) as received or fired, (2) dry or moisture free, and (3) combustible, which is the fuel ash and moisture free.

102. Coal Classifications, Chemical and Physical. Table 23 indicates proximate analyses and chemical constituents (ultimate analyses) of coals which are of commercial value. The different kinds of coal indicated therein fall within the limits of classifications, based on rank, as defined by the ASTM Standards of 1937.

The ASTM rank classifications separate anthracites, semi-anthracites, and low-volatile bituminous and medium-volatile bituminous coals by percentage limits of fixed carbon and volatile matter when

TABLE 23

ANALYSES OF COALS OF THE UNITED STATES

(Abstracted from *U. S. Bureau of Mines Bulletin* 22)

Rank	State	County	Sample	Proximate Analysis, Coal as Received				Ultimate Analysis, Coal as Received					Heat Value by Calorimeter, Btu per Lb	
				Moisture	Volatile Matter	Fixed Carbon	Ash	Sulfur	Carbon	Hydrogen	Oxygen	Nitrogen	As Rec'd	Dry
Anthracite	Colorado	Gunnison	Mine	2.70	3.32	88.15	5.83	0.80	85.38	2.98	1.19	1.12	14,099	14,490
Anthracite	Pennsylvania	Lackawanna	Car culm	5.41	7.02	71.79	15.78	0.74	72.65	2.50	2.15	0.77	12,047	12,737
Anthracite	Pennsylvania	Schuylkill	Mine	2.80	1.16	88.21	7.83	0.89	84.36	1.58	1.91	0.63	13,298	13,682
Semi-anthracite	Colorado	Gunnison	Mine	4.86	6.96	81.87	6.31	0.81	79.66	3.43	3.00	1.93	13,468	14,155
Semi-anthracite	Pennsylvania	Sullivan	Mine	3.38	8.47	76.65	11.50	0.63	78.43	3.20	1.86	1.00	13,156	13,617
Semi-bituminous	Maryland	Allegany	Car run of mine	3.42	17.65	71.84	7.09	0.84	80.17	4.34	2.69	1.45	14,162	14,665
Semi-bituminous	Pennsylvania	Cambria	Car run of mine	3.51	16.82	73.04	6.63	0.94	80.70	4.17	2.79	1.26	14,279	14,800
Semi-bituminous	West Virginia	Fayette	Car run of mine	2.52	21.28	68.20	8.00	0.68	78.71	4.45	4.33	1.31	13,997	14,359
Semi-bituminous	West Virginia	McDowell	Composite mine	3.50	15.00	78.30	3.20	0.60	84.39	4.20	2.73	1.38	14,800	15,330
Bituminous	Alabama	Bibb	Mine	3.16	31.05	59.56	6.23	1.20	78.28	4.98	4.78	1.37	14,141	14,602
Bituminous	Colorado	Boulder	Car run of mine	18.68	34.88	40.45	5.99	0.55	57.46	3.99	12.18	1.15	10,143	12,472
Bituminous	Illinois	Franklin	Mine	11.50	26.70	52.67	9.13	0.60	65.40	4.01	8.28	1.08	11,410	12,893
Bituminous	Illinois	La Salle	6-in. lump car	12.39	36.89	41.80	8.92	3.02	61.29	4.47	8.01	1.00	11,399	13,012
Bituminous	Illinois	Sangamon	Car run of mine	16.00	32.41	37.82	13.77	4.05	53.89	3.77	7.63	0.91	9,940	11,833
Bituminous	Illinois	Williamson	Mine	7.88	31.20	49.89	11.03	2.99	66.50	4.26	6.30	1.04	11,741	12,746
Bituminous	Indiana	Green	Car run of mine	10.30	36.31	41.64	11.75	4.23	61.00	4.24	7.42	1.06	11,218	12,508
Bituminous	Iowa	Marion	Car run of mine	14.21	33.17	37.40	15.22	4.66	54.08	3.93	6.59	1.31	10,019	11,678
Bituminous	Kansas	Crawford	Car run of mine	4.99	32.68	49.36	12.97	4.28	67.34	4.43	4.91	1.08	12,242	12,884
Bituminous	Kentucky	Hopkins	Car run of mine	7.92	36.09	45.93	10.06	3.52	65.29	4.51	7.30	1.40	12,022	13,055
Bituminous	Ohio	Hocking	Mine	9.72	32.44	53.41	4.43	0.54	69.50	4.62	9.94	1.25	12,247	13,567
Bituminous	Oklahoma	Pittsburg	Mine	2.09	27.59	50.25	20.07	5.73	63.66	4.23	2.89	1.33	11,695	11,945
Bituminous	Pennsylvania	Fayette	Car run of mine	5.13	27.87	58.29	8.71	0.86	73.13	4.34	6.33	1.50	13,365	14,089
Bituminous	Pennsylvania	Westmoreland	Car run of mine	3.39	31.79	56.46	8.36	1.0[illegible]	74.42	4.69	6.70	1.39	13,699	14,178
Bituminous	Tennessee	Campbell	Car run of mine	5.38	34.54	53.03	7.05	0.99	72.41	4.90	7.55	1.72	13,048	14,788
Bituminous	Virginia	Lee	Car run of mine	4.06	34.93	56.28	4.73	1.20	76.59	4.87	7.31	1.24	13,826	14,411
Bituminous	West Virginia	Kanawha	Car run of mine	2.82	32.20	56.95	8.03	1.38	76.74	4.85	4.84	1.34	13,766	14,166
Bituminous	West Virginia	Mingo	Car run of mine	2.86	33.23	58.08	5.83	0.67	78.38	4.69	6.14	1.43	14,105	14,521
Sub-bituminous	Colorado	Boulder	Mine	19.15	30.82	44.27	5.76	0.25	56.38	3.80	13.58	1.08	9,616	11,893
Sub-bituminous	Washington	King	Mine	17.51	31.17	38.55	12.77	0.37	52.11	3.90	12.26	1.08	9,331	11,312
Lignite	North Dakota	McLean	Lump	35.96	31.92	24.37	7.75	1.15	41.43	2.54	9.96	1.21	7,069	11,038
Lignite	North Dakota	Williams	Car run of mine	38.92	25.54	30.15	5.39	0.48	39.34	2.56	12.63	0.68	6,739	11,034
Lignite	Texas	Wood	Car run of mine	33.71	29.25	29.76	7.28	0.53	42.52	3.04	12.13	0.79	7,348	11,084
Lignite	Wyoming	Sheridan	Mine	29.33	28.58	36.42	5.67	1.21	45.09	3.01	14.80	0.89	7,783	11,012

determined on a dry basis. High-volatile, *A*, bituminous coals are classified according to their fixed carbon and volatile matter on a dry basis and the heating value as existent with their original vein moisture. High-volatile, *B* and *C*, bituminous coals; sub-bituminous coals *A*, *B*, and *C* together with lignites are separated into different ranks on the basis of their Btu per pound heating value when holding their vein moisture.

Important physical characteristics of different kinds of coals are listed in the following paragraphs.

Anthracite coal is very hard, shiny-black, non-coking, very low in volatile matter, high in fixed carbon, and slow to ignite unless the furnace temperature is high; it requires a strong draft, burns with little or no flame, and is desirable where smokeless combustion is essential.

Semi-anthracite coal differs from anthracite in that it has more volatile matter, less fixed carbon, less luster, and it burns with longer and more luminous flames.

Semi-bituminous coal is the highest grade of bituminous coal; it burns with little smoke, is softer and contains more volatile matter and less fixed carbon than the anthracites, and has a tendency to break up into fine pieces.

Bituminous coals are "soft," have high percentages of volatile matter, burn with long yellow smoky flames, and vary greatly in their composition. The distinctions between bituminous coals of eastern and mid-continent origin are in their percentages of vein moisture and volatile matter.

Sub-bituminous coals, known also as black lignites, are low-grade bituminous coals, which do not have the woody structure that characterizes lignites. They are high in ash, moisture, and volatile matter, and they disintegrate when exposed to air.

Lignites are the first step in coal formation after peat; they have a woody clay-like appearance and high moisture content, and they disintegrate readily.

Caking or coking coals fuse together; non-coking coals do not fuse. High-ash coals are those having an ash content of more than 10 per cent.

103. Coal Sizes. In mining operations, the coal produced ranges in size from large lumps to fine dust. Generally the fine sizes contain more impurities and dirt due to segregation and consequently have a lowered heating value.

Commercial sizes of bituminous coals differ somewhat in different areas of production, but they may be listed as run of mine, lump of various sizes, nut, screenings, and slack. Anthracite coal is listed as large egg, stove, nut, pea, buckwheat, and rice.

104. Prepared Solid Fuels. Coke is the solid residue left after the destructive distillation of certain kinds of bituminous coals and the final residue in the distillation of petroleum oils. The physical and chemical properties of either coal coke or petroleum coke are dependent upon (1) the specific kind of coal or oil used, (2) the method by which the volatile materials are driven off, and (3) the subsequent exposure to moisture.

Coal cokes consist largely of the carbon and the ash of the coal, together with small amounts of volatile materials. Petroleum cokes have high carbon content, low amounts of volatile materials, and high heating values.

The analyses of cokes vary greatly, but the data of Table 24 indicate their general characteristics.

TABLE 24

ANALYSES OF COKE

Proximate Analysis as Received			Ultimate Analysis as Received		
Sample	Coal	Petroleum	Sample	Coal	Petroleum
Moisture	10.77	5.50	Carbon	73.16	87.01
Volatile matter	4.92	5.82	Hydrogen	1.18	3.91
Fixed Carbon	72.26	87.50	Oxygen	1.13	0.63
Ash	12.05	1.18	Nitrogen	0.92	0.85
Heating values as received, Btu per lb	11,160	15,000	Sulfur	0.79	0.92
Heating value dry, Btu per lb	12,506	15,886			

Certain fine sizes of coals mixed with some binding material, such as pitch, tar, etc., and pressed into small blocks are known as *briquettes*. The heating value of briquettes is entirely dependent upon the calorific values of the coal and the binder employed. The use of briquettes is a matter of convenience, but the costs of production have made them relatively expensive.

Pulverized coal ground to dust fineness has a wide application in power plants but at the present has little application in domestic heating, although some progress has been made in this field.

105. Fuel Oils. The chief constituents of fuel oils are carbon and hydrogen, which are united in a series of very complex hydrocarbon compounds. Other items in the analyses are oxygen, nitrogen, sulfur, and moisture. The last is very undesirable, especially when it forms an emulsion. Crude oils are never used for heating, but their analyses are not much different from those of their fuel distillates. Typical properties of a few crude and fuel oils are given by Table 25.

TABLE 25

ANALYSES OF TYPICAL AMERICAN OILS*

Location	Physical Properties				Chemical Properties				
	Gravity, API at 60 Deg F	Specific Gravity at 60 Deg F	Flash Point, Deg F	Burning Point, Deg F	C Carbon	H Hydrogen	O Oxygen	S Sulfur	Calorific Value, Btu per Lb
Calif. crude midway	16.3	0.957	172	210	86.58	11.61	0.74	0.82	18,613
Penn. crude	39.8	0.826	...	...	82.00	14.80	3.20†	...	17,930
West Va. crude	36.7	0.841	...	...	84.36	14.10	1.60†	...	18,400
Texas crude	22.3	0.920	142	181	84.60	10.90	2.87†	1.63	19,060
Okla. crude	25.3	0.903	264	286	87.93	11.37	0.19	0.41	19,650
Texas fuel	21.3	0.926	216	240	83.26	12.41	3.83	0.50	19,481
Penn. fuel	28.2	0.886	...	...	84.90	13.70	1.40†	...	19,200

* *Fuel Oil in Industry*, by Stephen O. Andros.

† Includes nitrogen.

In general the percentages of the principal components range as follows: carbon, 80 to 87; hydrogen, 12 to 15; and sulfur, 0 to 5. The specific gravities vary from 0.80 to 0.97 and the heating values from 18,500 to 20,000 Btu per lb.

Fuel oils are standardized in six grades, which are designated by numbers, five of which are listed in Table 26. Grades 1, 2, and 3 are known as light, medium, and heavy domestic fuel oils. Oils classified by the numbers 4, 5, and 6 are respectively light, medium, and heavy industrial fuel oils. Number 4 oil is not ordinarily listed in fuel-oil specifications.

106. Fuel-Oil Specifications. Fuel oils should be hydrocarbon oils free from grit, acid, and fibrous materials, and they should have limitations on their sulfur, silt, and moisture contents. Specifications for fuel oils should include the items of viscosity, specific gravity, flash point, burning point, pour point, calorific value, and water, silt, and sulfur contents.

TABLE 26

REQUIREMENTS FOR FUEL OILS*

(National Bureau of Standards CS 12-40 except gravity and heating value)

Grade†	Flash Point, Deg F		Pour Point, Deg F	Water and Sediment, Per Cent	Carbon Residue Per Cent	Ash, Per Cent	Distillation Temperatures, Deg F				Viscosity, Seconds				API Gravity, Deg	Approximate Heating Valve, Btu per Gal
							10 Per Cent Point	90 Per Cent Point		End Point	Saybolt Universal at 100 F		Saybolt Furol at 122 F			
	Min	Max	Max	Max	Max	Max	Max	Max	Min	Max	Max	Min	Max	Min		
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel	100 or legal	165	0‡	Trace	0.05 on 10% residuum§		410			560					38-40	126,000
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel	110 or legal	190	10‡	0.05	0.25 on 10% residuum¶		440	600							34-36	138,700
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel	110 or legal	230	20‡	0.10	0.15 straight			675	600**		45				28-32	141,000
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel	130 or legal			1.00		0.10						50	40		18-22	148,500
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel	150			2.00‡‡									300	45	14-16	152,000

* Recognizing the necessity for low sulfur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulfur, max. per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No limit
No. 6	No limit

Other sulfur limits may be specified by only mutual agreement between the buyer and seller.

† It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

‡ Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

§ For use in other than sleeve-type blue-flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

‖ The maximum end point may be increased to 590 F when used in burners other than sleeve-type blue-flame burners.

¶ To meet certain burner requirements the carbon-residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

** The minimum distillation temperature of 600 F for 90 per cent may be waived if API gravity is 26 or lower.

‡‡ Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

Viscosity is a measure of the internal friction of the oil or its resistance to free flowing. Viscosity is specified at definite temperatures as determined by some specific viscosimeter and is the time in seconds required for a definite quantity of the oil, at the prescribed temperature, to flow through the orifice of the viscosimeter.

Specific gravity is the ratio of the weight of a given volume of the oil to the weight of an equal volume of water at the same temperature. A temperature of 60 F is commonly used so that sp gr 60°/60° F means that oil at 60 F has been referred to water at 60 F. Specific gravities may be determined by either a hydrometer or a Westphal balance. The oil industry uses the American Petroleum Institute (API) scale to express specific gravity in degrees. The API scale fixes a reading of 10 deg as equal to a specific gravity of 1.00. Readings greater than 10 deg indicate a specific gravity less than 1.0 or an oil which is lighter than water. Actual specific gravities may be calculated from degrees API by the following equation:

$$\text{Sp gr} = \frac{141.5}{131.5 + \text{deg API}} \tag{61}$$

Specific gravities are useful in computing the weights of unit volumes, as oils are usually sold either by the volume in gallons or barrels of 42 gal.

Flash point and *burning point* are, respectively, the temperatures at which, when an oil is heated, its vapors flash or burn momentarily and burn continuously when a small flame is brought near its surface. *Pour point* is the lowest temperature at which cold oil will flow.

Actual determinations of heating value, made with a calorimeter, are the most useful, as the calorific value may be only approximately calculated by Dulong's equation. Water and sediment contents should not exceed 0.05 per cent for No. 1 oil and 2 per cent by volume for No. 6 oil. Although sulfur is undesirable, an amount not in excess of 2 per cent is not seriously objectionable in the heavier oils.

107. Gaseous Fuels. The use of gas as a fuel in heating plants has become quite extensive especially since the advent of long-distance high-pressure transmission lines for the conveyance of natural gas. Data relative to various gases are given in Table 27.

Natural gas occurs in many localities and usually consists of a high percentage of methane, ethane, and small percentages of nitrogen, carbon monoxide, carbon dioxide, etc. The heat value of natural gas ranges from 850 to 1150 Btu per cu ft.

By-product coke-oven gases range from 400 to 600 Btu per cu ft in heat value and contain considerable percentages of hydrogen and methane, together with low percentages of carbon dioxide, nitrogen, oxygen, and carbon monoxide.

City gas is secured by the destructive distillation of a suitable coal and the decomposition of water vapor (steam). The product is often enriched by the use of a petroleum oil in the process.

TABLE 27

ANALYSES OF FUEL GASES

Gas	Composition by Volume, Per Cent								Heating Value, Btu per Cu Ft
	Hydrogen, H_2	Methane, CH_4	Ethylene, C_2H_4	Ethane, C_2H_6	Carbon monoxide, CO	Carbon dioxide, CO_2	Oxygen, O_2	Nitrogen, N_2	
Natural, Penn.	...	90.0	...	9.0	...	0.2	...	0.8	1126
Okla.	...	94.3	...	...	...	1.10	...	4.6	1006
Calif.	...	59.2	...	13.9	...	26.2	...	0.7	889
Coke-oven	50.0	36.0	4.0	...	6.0	1.5	0.5	2.0	603
	57.4	28.5	2.9	...	5.1	1.4	0.5	4.2	536
City gas	42.84	8.01	9.10	6.29	21.0	3.60	0.9	8.26	540

Liquid-petroleum gases, by-products of oil refineries, consist of butane, C_4H_{10}, and propane, C_3H_8. The materials can be liquified at reasonable temperatures and pressures which allow them to be transported in tank cars and stored in suitable containers on the purchaser's premises. The gases are used in various ways for both industrial and domestic purposes. Each may be used alone when vaporized and mixed with air, they may be blended together and mixed with air to form a fuel gas, or they may be mixed with other fuel gases, in various proportions, to augment a fuel supply. The higher heating value of butane is 21,320 and of propane 21,620 Btu per lb.

108. Calorific Value of Fuels. The heat liberated by the complete and rapid oxidation (or burning) of a unit weight or volume of a fuel is the *heating* or *calorific value* of the fuel. For solid and liquid fuels the heating value is expressed in Btu per lb. For gaseous fuels the heating values are expressed in Btu per cu ft of gas measured at a temperature of 60 F and a pressure of 30 in. of mercury.

Fuels which contain hydrogen or hydrocarbons have two heating values, the higher and the lower. *The higher heating value* of a fuel is the one indicated by any sort of a fuel calorimeter. *The lower heating* *value* is the net heat liberated per pound of fuel after the heat necessary to vaporize the steam formed from the hydrogen, as well as that from the fuel, has been deducted.

The heating value of a fuel may be found by two methods: by calculation and by the use of a fuel calorimeter. When the ultimate analysis of the fuel is known the heating value may be computed. The most satisfactory method of determining the heating value is to burn a quantity of the fuel in a calorimeter and to measure the rise in temperature induced in a given quantity of water. When the proper corrections are applied for the calorimeter parts, radiation, etc., the heat value per unit of the fuel may be determined from the calorimeter data.

109. Heating Value of a Fuel by Calculation. Dulong's formula for the heating value F of a solid fuel based on the actual weight of the total carbon C, the available hydrogen $H - O/8$, and the sulfur S is expressed in Btu per pound as

$$F = 14{,}540C + 62{,}000(H - O/8) + 4050S \qquad (62)$$

In the foregoing equation the values of C, H, O, and S for the fuel are obtained from its ultimate analysis and are expressed decimally. Dulong's equation gives only approximate results when used to calculate the heating value of a liquid fuel from its ultimate analysis.

The heating values of gaseous fuels may be calculated from their volumetric analyses if the calorific values per cubic foot of the various constituents are known. Such calculations are based on the assumption that each constituent is free and does not influence the others. The calorific values of several constituents of gases are given in Table 28.

TABLE 28

CALORIFIC VALUES OF GASES

At 60 F and 29.92 in. of mercury

Gas	Symbol	Heating Value, Btu per Cu Ft		Gas	Symbol	Heating Value, Btu per Cu Ft	
		Higher	Lower			Higher	Lower
Carbon monoxide	CO	322	322	Methane	CH_4	1008	916
Hydrogen	H_2	330	282	Ethylene	C_2H_4	1598	1510
				Ethane	C_2H_6	1763	1625

110. Combustion. This process is one of oxidation, the necessary oxygen being supplied from air. The burnable parts of the fuel ultimately become parts of the gaseous products of combustion. The prerequisites of good combustion are a furnace of proper design, an adequate supply of oxygen secured from a sufficient but not excessive air supply, a thorough mixture of the oxygen and burnable parts of the fuel, and furnace temperatures high enough to gasify the fuel, to ignite the mixtures, and to maintain the process.

Actually in the ordinary furnace fuels cannot be burned with the theoretical amount of air necessary for their combustion. Because of the limitations in getting oxygen to the combustible parts of a fuel it is necessary to supply some excess air in combustion processes in order to secure a better operating efficiency. Care must be taken to hold the amount of excess air to a minimum as it lowers the temperature of the products of combustion and causes greater heat losses in the dry flue gases by virtue of their increased weight over those formed with the proper air supply. The complete combustion of pure carbon with the theoretical amount of air would give a gas having about 21 per cent of carbon dioxide, CO_2. Most fuels contain both carbon and hydrogen; hydrogen burns to water vapor. Therefore such fuels never produce dry flue gases which show a percentage of CO_2 as high as 21. Excess air leads to a further reduction of the CO_2 volume in dry gases. For coals good combustion exists in domestic heating plants when the CO_2 of the flue gases ranges from 10 to 15 per cent; with oils 8 to 12 per cent CO_2 is satisfactory; and with gases containing hydrogen from 7 to 10 per cent CO_2 is practical.

Smoke is a term applied to escaping flue gases which are colored by fine ash particles, free-carbon particles, or both. Smoke may range in color from a slight gray haze to a deep black. Smoke resulting from incomplete combustion arises from the inability to maintain the combustion process under the conditions designated in the first paragraph of this article. Black smoke, with unburned free carbon particles, is indicative of poor combustion, and the greatest heat losses, occasioned by it, are due to unburned carbon in the form of carbon monoxide, CO, and unburned hydrogen and hydrocarbons.

111. Flue-Gas Analyses. Determination of the percentages by volume of the various constituents in the dry gases comprises a flue-gas analysis. Usually analyses include only the percentages of carbon dioxide, oxygen, carbon monoxide, and nitrogen. More complete analyses include the percentages of hydrogen and hydrocarbons that may be present. Flue-gas analyses should always be made of representative samples properly collected and protected. A portable analyzer, known as the Orsat apparatus, is shown by Fig. 39. The essential parts are: the measuring burette *A*, the leveling bottle *F*, *the gas cleaner H, and the absorption pipettes, B, C, D, and E* which are interconnected by means of a manifold having the ground cocks *I*. The measuring burette is water jacketed to prevent temperature and density changes of the gas. Carbon dioxide is absorbed in pipette *B*, which is partially filled with caustic potash KOH; oxygen is taken out in pipette *C*, which contains an alkaline solution of pyrogallic acid, and carbon monoxide is absorbed by an acid solution of cuprous chloride in pipettes *D* and *E*. The absorptions must occur in the order indicated, and the nitrogen is obtained by difference.

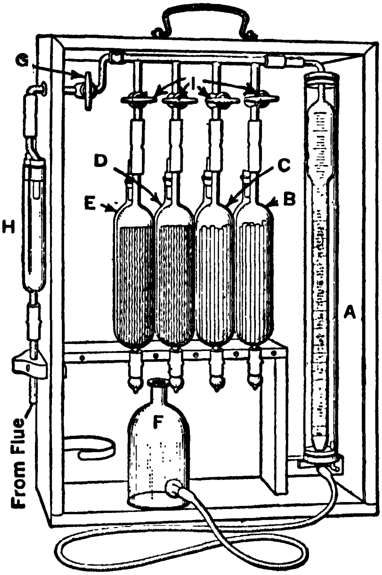


FIG. 39. Orsat apparatus.

112. Combustion Air and Flue-Gas Weights. All the following equations[1] may be derived in a logical manner. In each equation where they are used, W_f, the weight of fuel fired, and W_r, the weight of ash and refuse, are expressed in pounds. Weights of air W_{ta} and W_{aa} as well as the weight of dry flue gases W_{dg} are in pounds per pound of fuel fired. All constituents such as C_f the carbon; H, the hydrogen; O, the oxygen; and S, the sulfur per pound of fuel are expressed decimally as well as C_r the carbon per pound of ash and refuse. In the flue-gas analyses carbon dioxide, CO_2; carbon monoxide, CO; oxygen, O_2; and nitrogen, N_2, are percentages.

Theoretical weight of air,

$$W_{ta} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S \qquad (63)$$

Actual weight of air,

$$W_{aa} = \left[\frac{28N_2}{12(CO_2 + CO) \times 0.769}\right]\left[\frac{W_fC_f - W_rC_r}{W_f}\right] \qquad (64)$$

---

[1] *Steam, Air, and Gas Power*, Fourth edition, by Severns and Degler, John Wiley and Sons, Inc., New York, 1948. Chapter V, Steam Fuels and Combustion.

Excess air, per cent,

$$E_a = \frac{(W_{aa} - W_{ta})100}{W_{ta}} \tag{65}$$

Weight of dry gases,

$$W_{dg} = \left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right]\left[\frac{W_fC_f - W_rC_r}{W_f}\right] \tag{66}$$

113. Heat Utilization and Losses in Combustion Processes. Analyses of the performances of steam and water boilers and warm-air furnaces often include a study of heat utilization and losses in combustion processes involved during their use. The analysis is designated as a heat balance and accounts for the disposition of the heat of a unit amount, pound or cubic foot, of the fuel used. Items covered in the following discussion are for a solid fuel; modifications of the procedure can be made to cover both gaseous fuels and liquid fuels when necessary. The quantity of heat included in each item can be expressed as a percentage of the total heat per fuel unit.

Enthalpy of the fuel moisture h'', Btu per pound, appears as superheated steam in the flue gases leaving the boiler or furnace smoke outlet at their temperature t_g and a pressure of one psia, and h_f' is the enthalpy of the fuel moisture at the fuel temperature t_f. The weight of moisture M_a is the quantity of water vapor in the room air supplied for combustion at its temperature of t_a and M_f is the moisture in a unit of the fuel used. All other symbols are as defined in Arts. 109 and 112. All weights are expressed in pounds. The following tabulation accounts for the disposition of heat per unit of fuel used.

1. Heat absorbed, h_1 = weight W of the absorbing medium steam, water, or air times its enthalpy change, i.e., $W(h - h_{f1})$, as in equations 92 and 94 of Chap. 7.

2. Moisture in the fuel, $h_2 = M_f(h'' - h_f')$ (67)

3. Combustion of hydrogen, $h_3 = 9H(h'' - h_f')$ (68)

4. Moisture in air, $h_4 = 0.45M_a(t_g - t_a)$ (69)

5. Dry chimney gases, $h_5 = 0.25W_{dg}(t_g - t_a)$ (70)

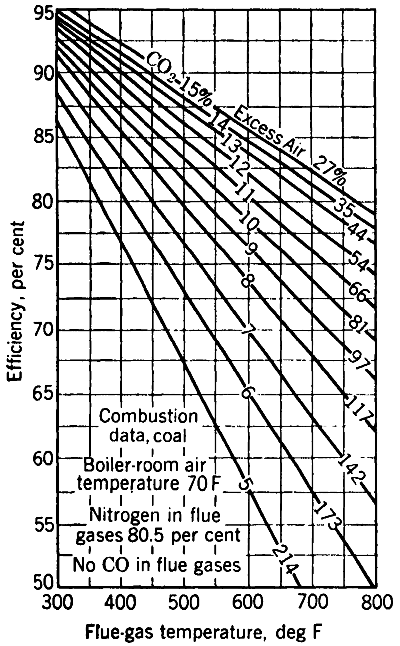
6. Incomplete combustion, $h_6 = 10{,}160\left[\frac{CO}{CO_2 + CO}\right]\left[\frac{W_fC_f - W_rC_r}{W_f}\right]$ (71)

7. Combustible in the ash and refuse, $h_7 = \frac{14{,}540W_rC_r}{W_f}$ (72)

8. *Radiation and unaccounted losses,* h_8 = *the calorific value F,* Btu per pound or cubic foot, of the fuel minus the sum items 1 to 7 inclusive. When incomplete combustion of hydrogen and hydrocarbons exists the losses so occasioned appear in this item unless more complete analyses are made than indicated by the procedure of Art 111. If the flue gases contain appreciable quantities of gases given in Table 28 the losses of heat due to their presence may be computed from their calorific values and included as separate items of the heat balance.

The ratio of item 1, of the heat balance, to the calorific value *F* of the fuel is an indication of the efficiency of heat utilization by a boiler or a warm-air furnace. Care should always be taken in the operation of the equipment to make the numerical value of item 1 as great as is possible. In the foregoing analysis, items 2 to 8 inclusive represent amounts of heat which are lost.

Losses of heat due to moisture produced by the combustion of hydrogen and hydrocarbons together with heat losses due to moisture in the fuel and the combustion air are generally of small magnitude. Heat losses due to the formation of carbon monoxide, CO, unburned hydrocarbons, and carbon in the ash and refuse can be reduced or minimumized by proper design of a furnace, control of the amount of combustion air, manipulation of the fire, and maintenance of clean heat-absorbing surfaces. The latter is of especial importance in the matter of heat absorption and the reduction of the temperature of the flue gases as they pass from the heating surfaces.



(*Copyright Industrial Press, 1934*)

FIG. 40. Boiler-efficiency chart.

The heat lost in the dry flue gases leaving the smoke outlet is dependent upon their temperature and weight. Figure 40 indicates boiler efficiencies possible when coal is used as a fuel to produce products of combustion of the temperatures and CO_2 and excess air contents as shown. This graph is based on no CO, hydrocarbon, moisture in coal and air, unburned carbon in the ash and refuse and radiation losses. The effects of excess combustion air are shown by the lowered percentages of CO_2. Similar graphs may be drawn for the results obtainable when either oils or fuel gases are burned.

114. Draft Requirements. Draft is the difference between the pressure of the air outside and that of the gases within a furnace, a boiler setting, or a chimney. Such pressure differences are variable in amount and are ordinarily of comparatively small magnitude. The measurements are usually made in terms of the height in inches of a water column exerting an equivalent pressure. Gages suitable for draft measurements are shown by Figs. 41 and 42. A light oil is used as the gage fluid, and the scale divisions are so placed that the readings obtained are in inches of water equivalent. Draft is required to produce the flow of the air necessary for combustion through the fuel bed and to promote the flow of gases from the furnace through the boiler setting and smoke connections into and out of the chimney or stack. Anything which offers resistance to the flow of air into and through the fuel bed and to the flow of flue gas from the furnace to the chimney increases the amount of draft that must be available. No fixed amount can be applied to all installations as each heating plant must be considered alone.

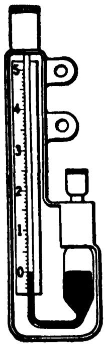


FIG. 41. Ellison vertical draft gage.

The Code ratings, Chap. 6, of hand-fired, solid-fuel-burning, gravity-flow, warm-air furnaces place a limitation of 0.08 in. of water on the draft losses between their ashpits and smoke outlets. Figure 116 shows that the draft requirements of one particular cast-iron, sectional, heating boiler range from 0.02 to 0.12 in. of water. As is usual with solid-fuel-burning furnaces, either hand or stoker fired, the necessary draft increases as the combustion rate with a given fuel becomes greater. In either of the cases cited the chimney must produce enough additional pressure differential to insure the flow of flue gases from the smoke outlet, through the breeching connection, into the chimney.

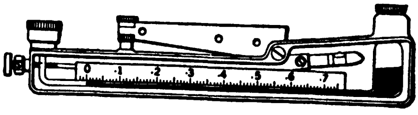


FIG. 42. Ellison inclined draft gage.

When a chimney is capable of furnishing a draft greater than that actually required, a cross damper placed in the breeching close to the operating unit can be utilized to give the necessary adjustment. Furnaces equipped to burn gaseous fuels usually operate with a pressure in the combustion chamber from 0.01 to 0.02 in. of water less than that of the outside atmosphere. Such furnaces should be fitted with draft diverters or hoods, Fig. 51, to prevent adverse operation due to wind action forcing a downward flow of air or gases in the chimney and its connections. A draft diverter prevents the pilot flames of gas burners from being extinguished.

The foregoing discussion is pertinent to heating boilers and furnaces; it is also applicable to power boilers, where the draft requirements may be much greater. For coals used as a fuel the data of Table 29 may be of some assistance in estimating furnace-draft requirements with different rates of combustion. The data do not include other draft necessities.

TABLE 29

TYPICAL CONDITIONS IN NATURAL-DRAFT FURNACE

Draft required in furnace for various rates of combustion

Combustion Rate per Square Foot of Grate, Pounds per Hour	Draft, Inches of Water	
	For Bituminous	For Anthracite
5	0.02 to 0.04	0.06 to 0.16
10	0.05 to 0.10	0.16 to 0.40
15	0.07 to 0.15	0.30 to 0.75
20	0.10 to 0.25	0.45 to 1.25
25	0.12 to 0.32	0.65
30	0.17 to 0.45	0.90

115. Natural Draft. The pressure difference thus secured is produced solely by the use of a stack or chimney. The magnitude of the force of draft is dependent upon the height of the chimney top above the level of the furnace grates, and the average difference between the temperature of the flue gases within the chimney and that of the outside air. Weather variations, boiler- and furnace-operation conditions, and the chimney height have a marked effect upon the amount of draft that may be secured by means of a particular chimney.

116. Chimneys and Stacks. Either of these structures is used with all draft systems for conveying the smoke or flue gases to an elevation above that of surrounding objects and buildings. Structures for the disposal of flue gases or the production of natural draft, if constructed of brick, concrete, or radial brick, are known as chimneys; those built of sheet steel or steel plates are designated as stacks. Stacks built of the lighter sheet metal require supporting guys; those built of heavier steel plates may be made self-supporting.

117. Chimney Height and Area. The head available for producing gas flow and overcoming frictional resistances, pounds per square foot, is equal to the chimney height, h ft above the grate level, times the difference between the density of the outside air, d_a, and the average density of the flue gases within the chimney, d_g. This pressure difference is $h(d_a - d_g)$. The pressure equivalent of one inch of water in pounds per square foot is $d_w/12$ when d_w is the weight of one cubic foot of water at the draft-gage fluid temperature. The theoretical draft, inches of water, is $h_w = 12h(d_a - d_g) \div d_w$. The density of the outside air and the average density of the flue gases may be computed by means of the gas-law equation $PV = MRT$. The density of either the air or the gas is equal to M when V is one cubic foot. The value of R may be taken as 53.35 ft-lb per deg F abs for both air and flue gases. When the barometric pressure B, inches of mercury, is known, $P/R = 144 \times 0.491B/53.35 = 1.326B$, when d_a and d_g are equal to $1.326B/T_a$ and $1.326B/T_g$, T_a and T_g being the absolute temperatures of the air and the flue gases respectively. Hence

$$h_w = \frac{15.91Bh}{d_w}\left(\frac{1}{T_a} - \frac{1}{T_g}\right) \tag{73}$$

For the required theoretical amount of draft, h_w in. of water, the chimney height in feet must be

$$h = (h_w d_w) \div 15.91B\left(\frac{1}{T_a} - \frac{1}{T_g}\right) \tag{74}$$

The theoretical amount of draft is seldom produced by a chimney, and the actual draft may range from 0.75 to 0.85 of that theoretically possible. For most calculations the actual draft may be taken as 0.8 of that theoretically possible. The theoretical draft h_c expressed in feet head of flue gases is $h(d_a - d_g) \div d_g$, or $1.326BhT_g(1/T_a - 1/T_g) \div 1.326B$, which simplifies to $h_c = h(T_g - T_a)/T_a$.

The theoretical velocity of gas flow is $v = \sqrt{2gh_c}$ fps. In terms of chimney height and temperatures

$$v = 8.02\sqrt{h(T_g - T_a)/T_a} \tag{75}$$

Under actual conditions of operation the flue-gas velocities may be only 30 to 50 per cent of the theoretical velocities owing to roughness of the interior surfaces of the chimney and leakages. The cross-sectional area of the chimney, square feet, is

$$A = Q \div vK \tag{76}$$

where Q = volume of the gases handled, cfs.
v = theoretical velocity of the gases, fps.
K = coefficient of velocity, 0.3 to 0.5.

The chief weakness in the foregoing method of design is in the assumption of the coefficient of velocity and the ratio of the actual draft to that theoretically possible.

Recommended sizes of chimneys and chimney heights for heating plants are given by Table 30 and Fig. 43.

TABLE 30

RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES*

Warm-air Furnace Capacity in Square Inches of Leader Pipe	Steam Boiler Capacity, Sq Ft of Radiation	Hot Water Heater Capacity, Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining, In.	Rectangular Flue		Round Flue		Height in Feet above Grate
				Actual Inside Dimensions of Fire Clay Lining, In.	Actual Area, Sq In.	Inside Diameter of Lining, In.	Actual Area, Sq In.	
790	590	973	8½ × 13	7 × 11½	81	...	. .	35
1000	690	1,140	...	...	...	10	79	...
...	900	1,490	13 × 13	11¼ × 11¼	127	...	...	.
...	900	1,490	8½ × 18	6¾ × 16¼	110	...	...	...
...	1,100	1,820	...	...	...	12	113	40
...	1,700	2,800	13 × 18	11¼ × 16¼	183	...	...	. .
...	1,940	3,200	...	...	...	15	177	. .
...	2,130	3,520	18 × 18	15¾ × 15¾	248	...	...	...
...	2,480	4,090	20 × 20	17¼ × 17¼	298	...	...	45
...	3,150	5,200	...	...	..	18	254	50
...	4,300	7,100	...	...	.	20	314	...
...	4,600	7,590	20 × 24	17 × 21	357	...	...	...
...	5,000	8,250	24 × 24	21 × 21	441	...	...	55
...	5,570	9,190	...	24 × 24†	576	...	...	60
...	5,580	9,200	...	...	. .	22	380	...
...	6,980	11,500	...	...	...	24	452	65
...	7,270	12,000	...	24 × 28†	672	...	...	...
...	8,700	14,400	...	28 × 28†	784	...	...	...
...	9,380	15,500	...	...	...	27	573	...
...	10,150	16,750	...	30 × 30†	900	...	...	...
..	10,470	17,250	...	28 × 32†	896	...	...	...

* Copyright, American Society of Heating and Ventilating Engineers. From ASHVE Code of Minimum Requirements for the Heating and Ventilation of Buildings, Edition of 1929.

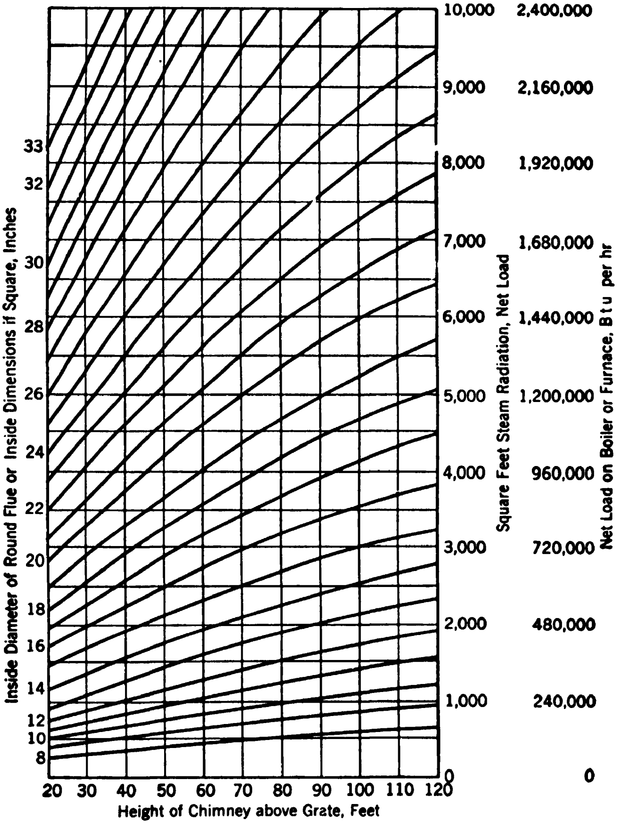
† Dimensions are for unlined rectangular flues.

Data relative to the commercial sizes of molded refractory flue linings are stated in Table 31.

When the chimney must have larger inside dimensions than are obtainable with commercial flue linings, fire bricks are used in the interior surfaces for some distance above the smoke connection to the chimney.

118. Important Items in Chimney Construction. Among other things the action of a chimney has a very distinct bearing on the capacity that a heating boiler or burner will develop. A plant otherwise properly designed will fail to give satisfaction if the chimney cannot produce the draft necessary to burn the required amount of fuel and carry away the gaseous products of combustion.

A chimney must not only be of adequate cross-sectional area and height but it must also be tight. Air leakage into the chimney will reduce the draft. There must be only a single smoke connection to the chimney, and boilers having smoke pipes attached to a common breeching must have dampers provided at their smoke outlets. The chimney should be straight, without offsets to catch soot and ash collections, and should extend at least 2 ft above the roof ridge or 3 ft above flat roofs. Account must be taken of the effect of adjacent higher portions of the building and nearby buildings and trees. When necessary the chimney height should be increased to offset the effect



(*Copyright The Industrial Press*)

FIG. 43. Chart for sizing natural draft chimneys.

TABLE 31

FIRE CLAY FLUE LININGS*

Actual Outside Size, Inches	Actual Inside Size, Inches	Actual Inside Area, Sq In.	Effective Inside Area, Sq In.	Weight, Lb per Ft
	Rectangular			
8½ × 8½	7¼ × 7¼	52.6	41.0	18.5
8½ × 13	7 × 11½	80.5	70.0	28.0
13 × 13	11¼ × 11¼	126.5	99.0	35.5
13 × 18	11¼ × 16¼	182.8	156.0	52.0
18 × 18	15¾ × 15¾	248.0	195.0	69.0
	Round			
7¼	6	28.3	28.3	12.0
9½	8	50.2	50.2	19.5
11¾	10	78.5	78.5	27.7
14	12	113.0	113.0	39.3
17⅛	15	176.7	176.7	54.3
20½	18	254.4	254.4	71.0
22¾	20	314.1	314.1	87.5
27¼	24	452.3	452.3	129.0
34¼	30	706.8	706.8	261.0
41	36	1,017.9	1,017.9	360.0

* Eastern Clay Products Association.

of downward-moving air currents produced by nearby objects and obstructions. Whenever possible the top of a chimney should be at least 35 ft above the grate level as chimneys of lower height are likely to be erratic in their action. Chimneys built in the interior of the structure are more effective than those exposed to the cold outside air. Circular cross-sectional areas are the most desirable, square ones being second in desirability. The effective area of a square chimney may be taken as the area of a circle which may be inscribed within its internal cross section. Rectangular chimneys offer greater frictional resistance to gas flow and allow greater losses of heat from the flue gases than those of other shapes. The ratio of the inside dimensions of rectangular chimneys should not exceed 2 to 1. Either fire brick or fire clay molded linings are desirable, and chimneys should be so constructed that in no way can they produce fire hazards.

119. Automatic Fuel-Burning Equipment. The objectives to be obtained by the use of automatic fuel-burning equipment are: (1) to utilize the fuel available with the maximum efficiency of combustion through the uniformity of fuel feed and the control of the combustion air, (2) to make use of fuels which require special equipment, (3) to reduce the manual labor and attention given to firing operations, (4) to maintain uniform temperature conditions in the spaces to be heated, and (5) to secure cleanliness of operation.

Included in the group of appliances are: mechanical stokers for the combustion of coals and cokes, oil burners, and gas burners. These units serve heating boilers and warm-air furnaces designed especially for their use; heating units, built for hand firing, which have been adapted to their use by the installation of either stokers or conversion oil and gas burners; and those heating boilers and warm-air furnaces designed especially for the use of either oil or gas as a fuel and in which the burners are made an integral part of them at the time of their construction.

120. Automatic Stokers. Large central heating plants may generate steam, dependent upon the coal available, with the aid of stokers of the types and sizes suitable for commercial power plants. Domestic heating plants employ stokers of either the overfeed or the underfeed types, with the majority of the installations being of the latter classification.

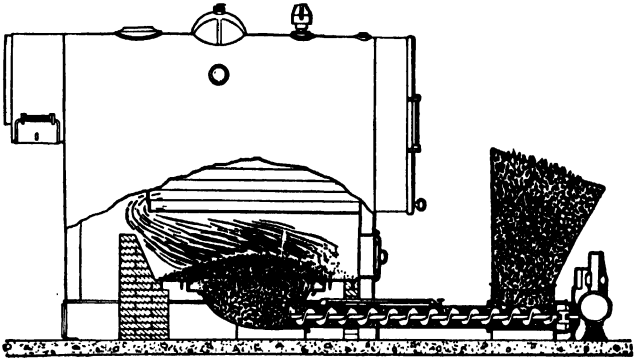


FIG. 44. Underfeed stoker with heating boiler.

Overfeed stokers include spreader forms with flat and inclined grates; traveling grates with fuel feed from a hopper at the front of the grate; and inclined rocking grates with the fuel coming to them from a hopper at the upper edge of the grate. The burning characteristics of the coal used have a bearing in their selection as a coal with a low-temperature-fusing ash cannot be employed. Furthermore such stokers are not adaptable to small-size installations.

Single-retort underfeed stokers can be constructed for small-size heating units, and when they are used fusible ash can be readily removed from a furnace in the form of a clinker. These stokers have a limited use with free-burning coals which do not have an ash which melts readily. Under such conditions the mechanism may include provision for the automatic removal of the fine ash and refuse material.

A typical single-retort underfeed stoker is illustrated by Fig. 44. The unit has a worm screw to convey coal from the hopper to the retort where ignition of the fuel occurs. The screw feed through the coal tube is commonly used, but some units of this type employ a ram either mechanically or hydraulically driven. Stokers are also arranged so that the coal bin serves as a fuel hopper and the stoker automatically conveys the coal from that location to its retort. Underfeed stokers invariably have a blower which supplies air under pressure to the tuyères (air openings) located about the top of the retort which is placed in the furnace. Small stokers have circular basket-shaped retorts; those of larger size use a *U* or trough-shaped construction of this part. Coal forced upward through the retort burns after it passes the level of the air tuyères. Heat from the combustion process causes volatile materials from the fuel to be released by distillation in the lower part of the fuel bed. These gases are burned as they pass into a region of high temperature, and when the stoker is properly handled combustion is complete, and the formation of smoke is prevented.

The United States Department of Commerce together with the Stoker Manufacturers Association (SMA) have classified stokers according to their coal-feeding rates in pounds per hour. These classes and capacities are: I, under 61; II, 61 to 100; III, 101 to 300; IV, 300 to 1200; and V, 1200 and over.

121. Stoker Furnaces. The amount of furnace volume allowed for the process is of prime importance in any combustion operation. This is especially true in connection with stokers which may function with high fuel-burning rates. The Stoker Manufacturers Association recommends the following empirical equations for the setting height of a steel-boiler crown sheet above the dead plates at the top of the retort level when the amount of coal to be burned per hour is known. For coal burning rates up to 100 lb per hr

$$D = 0.1125B + 15.75 \tag{77}$$

and for rates from 100 to 1200 lb per hr

$$D = 0.03B + 24 \tag{78}$$

where D = minimum setting height, in.
B = coal-burning rate, lb per hr.

When cast-iron sectional boilers are used the height may be $\frac{7}{8}$ of D. The SMA also has recommendations for firebox dimensions for the different classes of stokers.[2]

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[2] *Technical Manual,* Stoker Manufacturers Association.

122. Stoker-Size Selection. Fuel-burning capacity is desired when a stoker is to be selected for a given installation, as it is desirable that conditions be maintained so that the fuel burns as fast as it is fed into the furnace. In order to find the required pounds of coal that must be burned per hour the following items must be known: total load, Btu per hour; calorific value of coal, Btu per pound; and the efficiency of the heat of the fuel utilization by the boiler or warm-air furnace. The total load for steam and water boilers is assumed to be 1.33 times all of the heat in Btu per hour required for any and every purpose. This assumption provides for an additional 33 per cent pickup allowance, and when the conditions are unusual the factor should be more than 1.33. An expression for the amount of fuel required per hour is

$$W_f = \frac{H}{F \times E} \tag{79}$$

where W_f = coal required per hour, lb.
H = total load, Btu per hr.
F = heating value of fuel, Btu per lb.
E = overall efficiency of unit.

123. The Control of Stokers. All automatic fuel-burning equipment requires, for its successful operation and the safety of equipment, the installation and satisfactory functioning of various controls and protective devices. Stokers for small heating units, such as those installed in residences, function with part or all of the following: (1) a room thermostat which starts and stops the stoker as needed, (2) a pressure-limit device for steam boilers, (3) water-temperature limiting equipment placed in the outlet of a hot-water boiler, (4) an air-temperature control placed in the bonnet of a warm-air furnace, (5) a low-water cutoff in the case of steam boilers, and (6) a hold-fire control to operate the stoker and to keep the fire alive when the weather is so mild that there is no demand for heat made by the controlling thermostat.

124. Oil Burners. The devices termed oil burners function to supply oil to, and to prepare it in some manner for combustion in, a furnace. Fuel oil cannot be burned as a liquid but must be gasified and mixed with the proper amount of air. The mixture requires ignition and a temperature high enough to insure complete burning of the oil vapor.

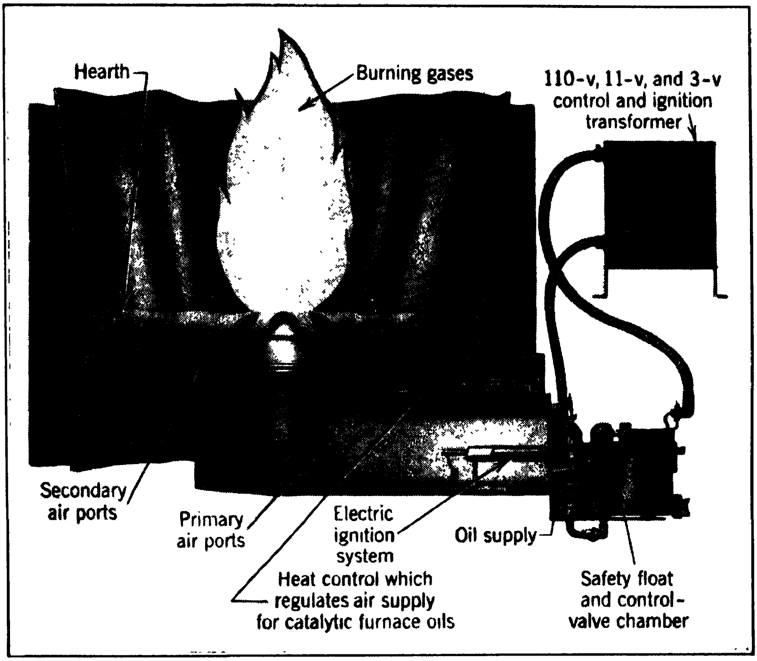
Fuel oils may be gasified by vaporization upon hot surfaces, or they may be atomized prior to vaporization. Atomization consists of breaking an oil up into very small globules, which greatly increases its exposed surface areas. Heat derived from either hot brickwork or burning oil in the furnace, acting upon the large amount of oil surface areas, produces the desired gasification. The oil vapor when mixed with sufficient air and held at the proper temperature undergoes combustion reactions to produce carbon dioxide. CO_2, and water vapor when completely burned. Care must be taken to provide a combustion space of sufficient volume in which the necessary temperatures for complete combustion may be maintained. The excess air supplied should be limited to a reasonable amount. Although 15 per cent of CO_2 is possible in the flue gases, when oil is used as a fuel, more satisfactory results are obtained if the amount of excess air supplied is such that the flue-gas analysis indicates from 10 to 12 per cent of CO_2. This adjustment can be gaged somewhat but not altogether by the flame color, which should be a bright orange.

Domestic oil-burner classifications may be made according to the air supply, the method of oil gasification, the type of flame produced, the scheme of fuel ignition as the burner starts functioning, and the manner of operation. Combustion air may be obtained by the action of natural chimney draft, by the operation of a fan or blower, or by a combination of the two, so that the primary air is supplied by the fan and the secondary air by the action of the chimney draft. Fuel oils may be atomized by centrifugal action obtained from a rotating disc or cup, by forcing the oil under pressure through a small orifice, by the use of either high-velocity steam or air jets in a special nozzle, or by forcing under pressure oil entrained with air through a nozzle. The burning oil may produce either a luminous or a non-luminous flame.

Ignition can be secured from a continuous high-voltage spark or a spark which functions as the burner starts operation. Gas pilot lights of various sorts and combinations of electric sparks and gas pilots are also used for the initial ignition as the burner starts operation. Burners may operate as follows: on and off, high or low flame, and a graduated flame according to the load to be handled. General classifications of oil burners are vaporizing or pot, rotary, and pressure-atomizing or gun types.

One form of vaporizing or pot-type burner is shown by Fig. 45. This unit operates with gravity feed of the oil and with natural draft furnished by a chimney capable of producing a force of 0.06 to 0.08 in. of water. As indicated in Fig. 45 oil flows over the bottom of the burner and is ignited electrically. The heated and vaporized oil is mixed with primary air which enters the burner through ports located above the oil heating and vaporizing level. Secondary air to complete the process of combustion is admitted through an opening between the upper and the lower burner ring. Combustion of the oil vapors which are mixed with the proper amount of air is completed in the furnace above the hearth level.

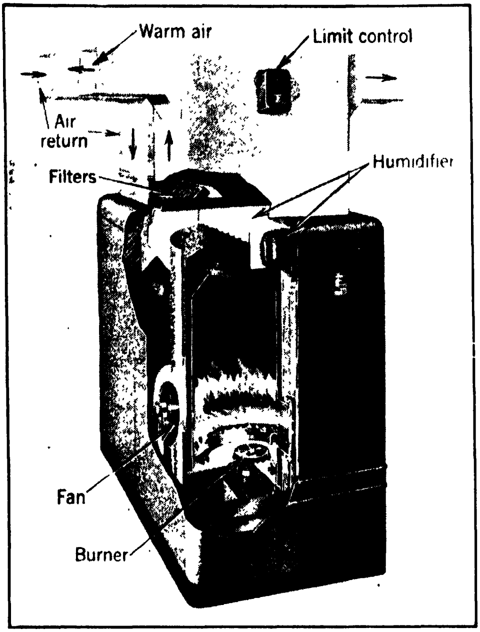
The rotary burner of Fig. 46 is included as an integral part of a winter air-conditioning furnace. Enlarged details of the burner parts are shown by Fig. 47 in which assembly *A* is a rotating unit which is comprised of oil-distributing tubes and a fan. The rotating element is located in the center of the combustion chamber with the operating mechanism sealed off from the furnace heat by a hearth made of refractory cement materials. By the action of the rotating portion of the burner of Fig. 47 droplets of oil are thrown outward toward the steel flame rim, placed around the edge of the hearth, and embedded within it. Ignition of the oil is by means of an electric arc which is shut off as soon as the flame rim becomes hot enough to maintain combustion. Oil vapors from the hot flame rim rise and are mixed with air supplied by the fan. Placed above the flame rim are grills which become hot and which are a further aid in mixing air and oil


Hearth
Burning gases
110-v, 11-v, and 3-v control and ignition transformer
Secondary air ports
Primary air ports
Electric ignition system
Oil supply
Heat control which regulates air supply for catalytic furnace oils
Safety float and control-valve chamber

(*H. C. Little Burner Co., Inc.*)

FIG. 45. Vaporizing or pot oil burner.

vapors and in maintaining the process of combustion. The flame rims with the attendant grills can either be circular or rectangular in shape to fit the furnace served.

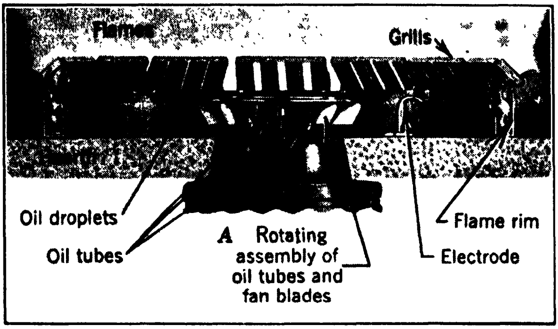


(*Timken Silent Automatic Division, Timken-Detroit Axle Co.*)

FIG. 46. Mechanical warm-air furnace with rotary wall-flame oil burner.

Figure 48 shows a gun-type oil burner suitable for conversion and other installations. In this particular unit, oil is atomized by passing it through a nozzle, and the combustion air is furnished by the action of a blower. Initial ignition of the oil is accomplished by the action of a spark across the terminals of the electrodes. Gun-type burners separate into two further classifications, which are low-pressure and high-pressure units. The low-pressure units operate with the oil being atomized when its pressure ranges from 2 to 7 psi; high-pressure units have the oil atomized at a pressure of approximately 100 psi.

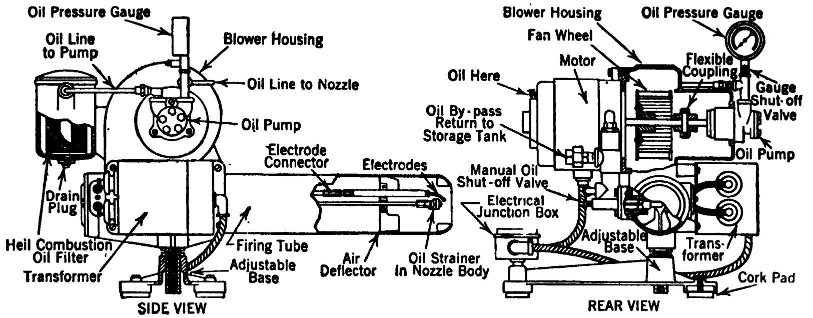
Automatic controls are important parts of any installation of oil burners. These devices which are discussed in Chap. 20 should



(*Timken Silent Automatic Division, Timken-Detroit Axle Co.*)

FIG. 47. Rotary wall-flame oil burner.

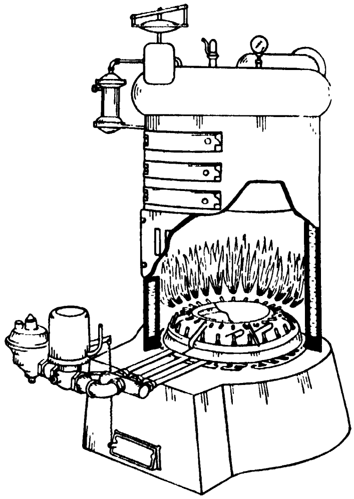
include a thermostat to control the burner to give the desired room temperature and safety devices to give protection against excessive boiler pressures or temperatures, low water, and ignition failure.



(*The Heil Co.*)

FIG. 48. Domestic gun-type oil burner.

The ratings of oil burners are based on the gallons of oil which they will burn per hour when installed as conversion units. For integral installations in boilers and warm-air furnaces the ratings of the combined units are usually given in Btu per hour.

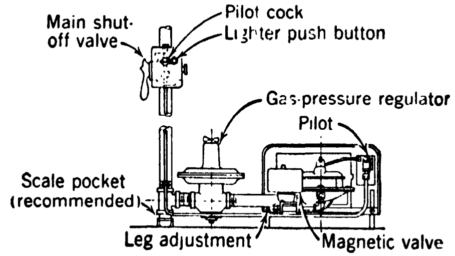


(*Barber Gas Burner Co.*)

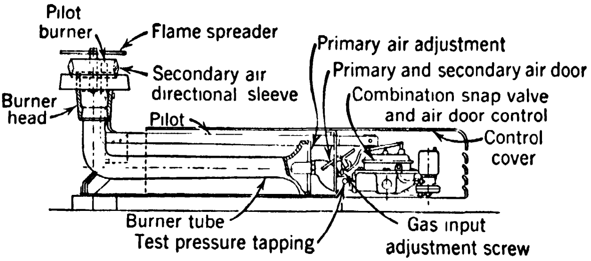
FIG. 49. Conversion gas burner.

125. Gas Burners and Gas-Fired Heating Units. A gas burner functions to convey a mixture of air and gas to the combustion chamber of a furnace where ignition and burning occurs. Gas burners operate with either atmospheric or power injection of combustion air. One form of gas burner is shown by Fig. 49, which is a conversion unit installed in a heating boiler. More details of a conversion burner suitable for heating boilers and warm-air furnaces are given by Fig. 50, which illustrates a unit which has a flat circular disc to direct the flames toward the furnace walls. The combined burner-boiler unit of Fig. 51 includes a draft divertor or hood which should be part of every gas burner installation. The divertor includes an inverted cone about which the flue gases pass on their way to the chimney. An annular opening in the divertor hood allows air from the furnace room to be drawn into the flue connection. The objectives of the draft divertor are to prevent a chimney from producing excessive draft conditions in the furnace and also to prevent downward flow of air in the chimney from extinguishing either a pilot flame used to ignite the gas flowing from the burner or the main burner flame.

Some conversion burners have refractory baffles to direct radiant heat toward the furnace walls, and they operate with non-luminous flames. Others operate without refractory baffles but have spreader plates or cones and produce luminous flames. In either case an attempt is made to produce the greatest amount of heat transfer to the heating medium through the surfaces surrounding the furnace as the transfer of heat is low in the flue passages of a gas-fired unit.



Front elevation


Pilot burner
Flame spreader
Primary air adjustment
Primary and secondary air door
Secondary air directional sleeve
Combination snap valve and air door control
Burner head
Pilot
Control cover
Burner tube
Test pressure tapping
Gas input adjustment screw

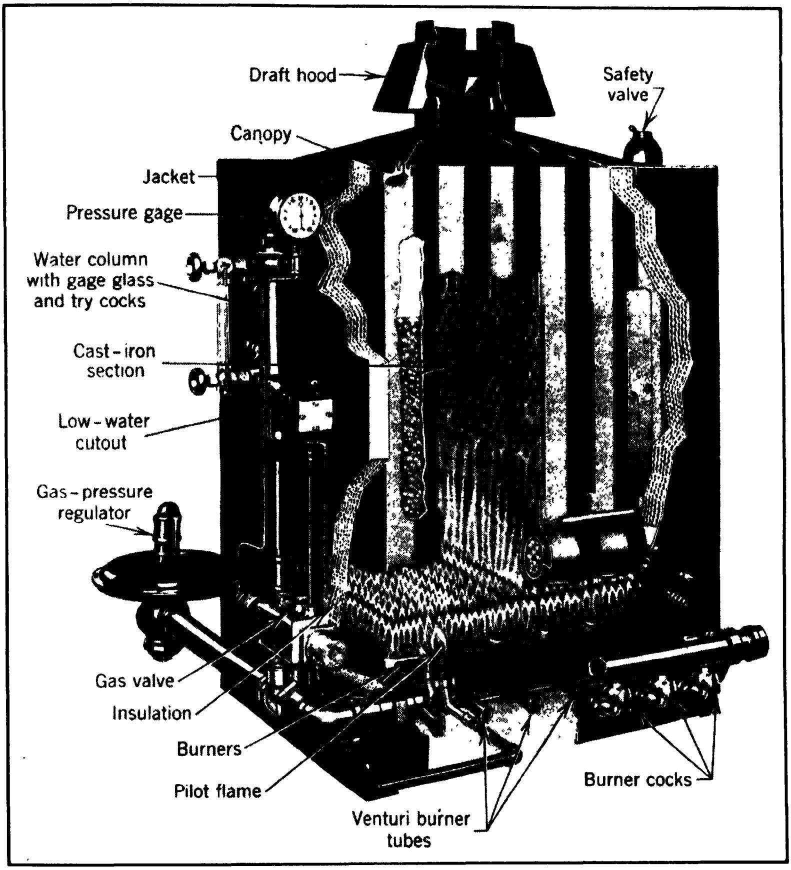
Side elevation

(*Bryant Heater Co.*)

FIG. 50. Spreader-flame conversion gas burner.

Gas burners are rated in terms of the hourly input of heat units to the furnace for the particular kind of gas burned.

The controls needed for gas-burner operation include room thermostat, gas-pressure regulator, thermostatic pilot valve in the gas supply line, pressure and temperature limit control, and other safety devices deemed necessary for stoker- and oil-fired boilers. Each gas burner should be fitted with automatic shutters which function to prevent the circulation of air through the heating unit to its chimney when the burner is not in operation.



(*American Radiator and Standard Sanitary Corp.*)

FIG. 51. Gas-fired steam-heating boiler.

PROBLEMS

1. A fuel oil has a specific gravity of 0.88 at 60 F and a calorific value of 142,000 Btu per gal. Find the API gravity and the heating value of the oil in Btu per pound.

2. A fuel oil has an API gravity of 21 deg at 60 F and a calorific value of 19,500 Btu per lb. Find the specific gravity of the oil and its heating value per gallon.

3. Calculate the higher heating value of a coke-oven gas which has the following analysis in percentages by volume: hydrogen, 55; methane, 32; ethylene, 3.0; carbon monoxide, 5.5; carbon dioxide, 1.5; oxygen, 0.5; and nitrogen, 2.5.

4. Calculate the heating value of one pound of McDowell County, West Virginia, semi-bituminous coal both on the basis of coal as received and as dry coal.

5. Calculate the heating value of one pound of Franklin County, Illinois, bituminous coal both on the basis of coal as received and as moisture- and ash-free coal.

6. Find the percentage of excess air used when a Green County, Indiana, coal is completely burned to give the following flue-gas analysis expressed as percentages: carbon dioxide, CO_2, 10.5; oxygen, O_2, 7.6; carbon monoxide, CO, 0; and nitrogen, N_2, 81.9.

7. How many pounds of dry flue gas are formed per pound of coal as fired in problem 6 if from each 1000 lb of fuel fired to a furnace 135 lb of ash and refuse are withdrawn with 15 per cent of carbon content?

8. The following data resulted from an evaporative test for a heating boiler. Water supplied at 180 F was evaporated into steam at 16 psia and 0.99 quality at the rate of 8 lb per lb of Crawford County, Kansas, coal as fired. The following temperatures represent average conditions: flue gases leaving the boiler, 500 F; room air, 80 F; and fuel, 80 F. The average flue-gas analysis was CO_2, 12; O_2, 6; CO, 0.5; and N_2, 81.5 per cent. The carbon in the ash and refuse amounted to 0.02 lb per lb of fuel fired and the moisture in the combustion air was 0.01 lb per lb of dry air. Calculate and tabulate a heat balance based on the coal as fired.

9. A chimney 75 ft high operated with flue gases at an average temperature of 450 F when the outside-air temperature was −5 F. Find the theoretical draft in inches of water and the probable velocity of gas flow in the chinney if the coefficient of velocity is 0.45.

10. A chimney 125 ft high receives flue gases at a temperature of 700 F. The average gas temperature in the chimney is 500 F when the outside-air temperature is 0 F. The coefficient of velocity may be taken as 0.35. Find the inside diameter of the chimney if it is to serve furnaces delivering 25,000 lb of dry flue gases per hr when the barometric pressure is 29.0 in. of mercury.

CHAPTER 6

HEATING WITH WARM-AIR FURNACES

126. Applications and Classifications. Furnace systems are suitable for buildings of relatively small size. Included in this category are residences, schools, stores, and churches having dimensions which will permit the circulation of air to and from a fuel-burning heater. The circulation can be produced either by gravity flow or mechanically. In any event the temperature of the warmed air delivered to the spaces to be heated should not exceed 175 F with gravity-flow and 150 F when the air is circulated by mechanical means.

127. Advantages of Warm-Air Furnace Heating. Warm-air furnaces are generally lower in first cost than steam and hot-water plants. They possess flexibility of operation as they begin to function quickly after a fire is started. The operating costs compare favorably with those of other systems. The temperature of the air entering rooms is adjustable to weather conditions, and recirculation of the air through the furnace is an aid in securing uniformity of air temperature. Humidification of air is more easily accomplished than in systems using radiators and convectors. Warm-air furnace systems take little occupied space above the basement, and there is no danger of damage to them due to freezing if left without a fire in winter.

128. Comparison of Gravity-Flow and Forced Circulation. Gravity systems are noiseless as far as the air circulation is concerned, and both the initial cost and the cost of maintenance are less with this type of warm-air system. However, gravity-flow systems have certain limitations which are not present when the air circulation is produced by means of a fan. Structures more than 40 ft square and above three stories in height are not suitable for gravity-flow heating. The furnace occupies considerable space in the basement and requires a central location because the leader pipes should be limited to lengths of 10 to 12 ft. Adequate spaces for stacks in the interior partitions may be more difficult to locate or not obtainable. Leaders may not run through unheated spaces unless exceptionally well insulated. The most effective air filters cannot be used for cleaning because their resistance to circulation is too much to be overcome by the small gravity head. Gravity systems require larger ducts for conveying the air to and from the furnace which interfere with the use of the basement for recreation or other purposes. Because of the lower velocity of the air circulation in gravity systems, heat transfer from the furnace to the moving air is not as good, and a greater portion of the heat absorbed from the furnace is lost between it and the warm-air registers. Forced-circulation systems which are equipped with devices for adding water vapor to the circulating air are often referred to as winter air conditioners.

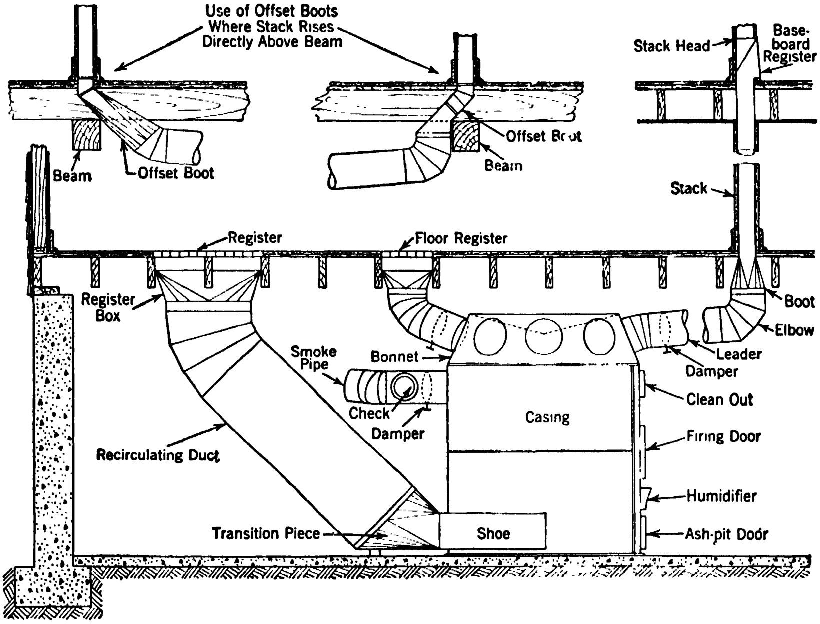


FIG. 52. Gravity-flow warm-air furnace system.

129. Installation and Action of Gravity-Flow Piped Furnaces. Examination of Figs. 52 and 53 indicates that the air to be warmed enters the casing at the bottom through the distributing shoe or boot which is joined to the recirculating duct leading from the heated space. The shoe should not extend above the level of the grate and must be wide enough so that its cross-sectional area is not less than that of the recirculating duct which serves it. Best operating results are to be obtained when the furnace can be served by a minimum number of short direct recirculating ducts designed to reduce the air-friction losses to a minimum.

The air entering the furnace casing is distributed from the shoe around the ashpit section and rises to pass over the heating surfaces of the furnace where it is warmed. The casing lining together with the air space between it and the casing reduces the losses of heat from the casing. The warmed air rises to the bonnet which serves as a plenum chamber from which the air is conveyed by the leader pipes toward the spaces to be heated. All leader-pipe collars should be attached to the bonnet with the tops of the openings at the same level, thereby preventing one leader from being unduly favored at the expense of the operation of other leaders. With the best arrangements, the leaders are taken radially from the bonnet, and an amount of space in the bonnet periphery proportional to the carrying capacity of each leader is allotted to it. The leaders pitch upward in the direction of air flow with a minimum uniform grade of one inch per foot. Leaders serving first-floor rooms are attached to either register boxes placed in the floor or boots and register boxes for baseboard units. For second- and third-floor rooms, stacks in the inside wall construction are necessary for the conveyance of the air to the upper floors. Each stack is usually rectangular in shape and requires a transition piece known as a boot at the point where the circular leader is joined to it. As shown in Fig. 52, the stack is fitted with a head suitable for either a baseboard or a wall-type register outlet. The best results are obtained if a stack has only a single register outlet.

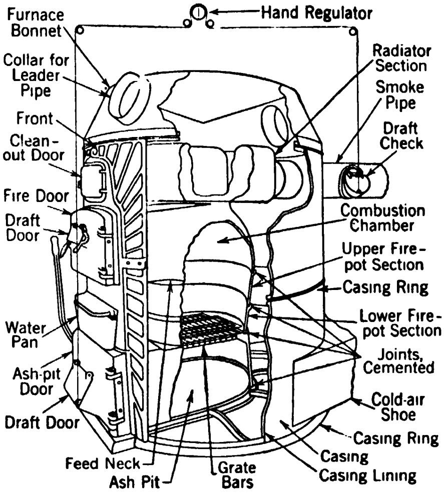


FIG. 53. Sectional cast-iron warm-air furnace.

130. Warm-Air Furnace Construction. Heating units, intended for the burning of solid fuels, are made of cast-iron sections having tongue and grooved joints, which are filled with heat-resisting cement, or they are made of steel plates which are either riveted or welded together.

Figure 53 illustrates a cast-iron sectional furnace having a ring radiator. The sectional elevation also includes a view of the galvanized sheet-steel casing, its lining, and the bonnet at the upper part of the casing. The names of the component parts are as indicated by Fig. 53.

One form of steel furnace is shown by Fig. 54. This unit has welded seams and, owing to the action of heat from the incandescent fuel, requires a refractory lining inside the firepot. The casing and bonnet arrangements are the same as those employed with cast-iron furnaces. Either of the two furnaces illustrated is suitable for use in both gravity-flow and forced-circulation systems.

In either furnace, the heating surfaces may be classified as direct and indirect. Direct surfaces are in contact with the fire or receive heat by radiation from the fire and are much more effective than the indirect surfaces which receive heat only from the hot gases. Some furnaces have projecting fins, etc., attached to portions of the unit, which serve as extended surfaces. The portions of such extensions farther than 3 in. away from the actual heating surface are not effective.

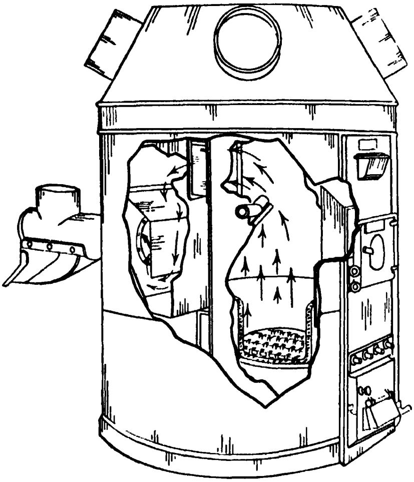


FIG. 54. Steel warm-air furnace.

Heat is transferred by convection to air moving over the heating surfaces. The rapidity of this transfer of heat is dependent upon the velocity of flow over the surface and the difference between the temperature of the air and that of the surface over which it flows. Radiant heat thrown out from the hot furnace surfaces is intercepted by the furnace casing lining which also becomes a convection heating surface. The rate of heat transfer per square foot of heating surface of furnaces, with gravity flow, averages somewhere between 2000 and 4500 Btu per hr with moderate combustion rates.

131. Performance of Gravity-Flow Warm-Air Furnaces. The curves of Fig. 55, taken from *Bulletin* 141 of the University of Illinois Engineering Experiment Station, indicate the performance of a hand-fired solid-fuel-burning piped furnace of a representative type. The construction of the furnace somewhat affects its performance under given conditions, but these figures, in general, are representative of gravity-flow warm-air furnace operation.

The efficiencies given are the ratio of the heat delivered at the bonnet to the heat supplied in the fuel. All heating plants, having boilers and furnaces located within the structure, operate so that the plant and the building function as a unit, and the overall efficiency for the two is greater than that based on the ratio of the boiler or heater output, at either the nozzles or bonnet, to the heat in the fuel used. The actual heat supplied to a structure is the total heat of the fuel consumed minus the sum of the heat lost in unburned fuel in the ash and refuse and the heat lost to the outside air by the chimney gases as they pass through the chimney and escape to the outside air. In a building having a chimney passing through its interior the heat of the fuel not put into the structure is the calorific value of the unburned fuel in the ashes and the heat in the flue gases in the chimney at the roof line. It is true that part of the heat thus accounted for is not placed directly in the spaces to be heated, but nevertheless it aids in warming the structure. When the heat of the fuel is thus accounted for, the overall efficiencies of operation for heating system and building range as high as 75 per cent with bituminous coal and 90 per cent with anthracite.

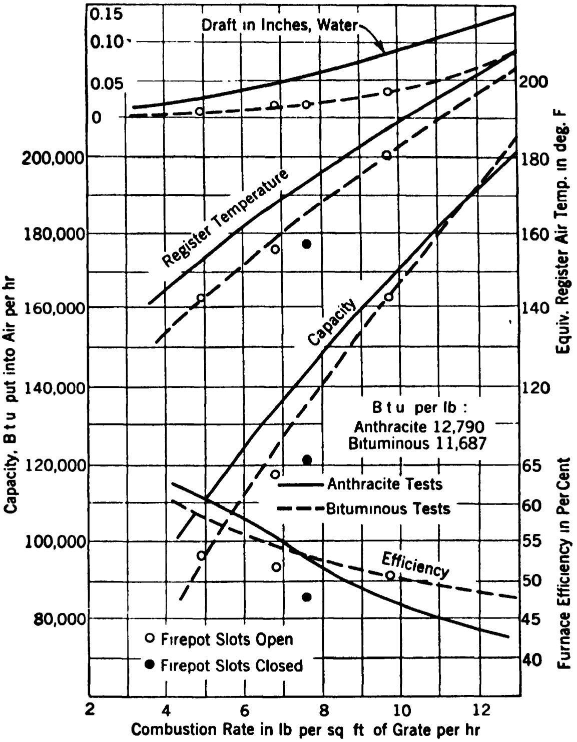


FIG. 55. Performance comparison for a cast-iron circular-radiator furnace, casing diameter 52 in.

In applying the curves of Fig. 55 the student should understand that the register temperature chosen fixes the draft required, the combustion rate, the furnace efficiency, and the capacity of a given furnace. The capacity curves shown by the graphs are true only for the particular furnaces for which they were drawn, as the capacity is also dependent upon the grate area. The drafts, register temperatures, efficiencies, and combustion rates shown are applicable to small furnaces of different amounts of grate area.

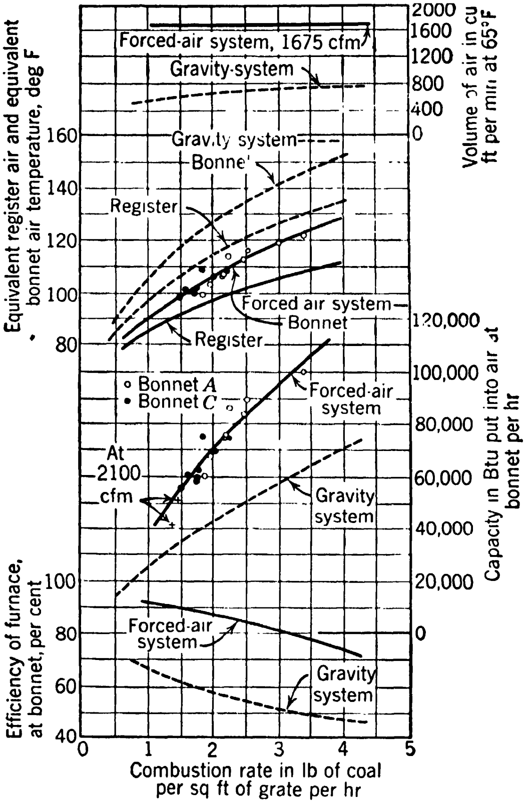


FIG. 56. Comparisons of operating results with gravity- and forced-circulation warm-air heating systems.

132. Performance of Forced-Circulation Warm-Air Furnaces. When the air is circulated through a warm-air furnace system by mechanical means, the temperature of the air delivered from the bonnet is determined by the capacity of the fan in proportion to the heat input to the furnace. Because of the greater volume of air circulated through the system when a fan is used, the temperature rise of the air is decreased, the register air temperature is decreased, and the efficiency of the furnace is increased. The curves shown in Fig. 56, from *Bulletin* 266 of the University of Illinois Engineering Experiment Station, were all plotted from data taken with the same furnace and give a direct comparison of the results obtained by forced circulation of the air and when the circulation was by gravity action. Because of the smaller ducts and the greater velocity used in a mechanical system the heat lost between the furnace and the registers is reduced, resulting in a greater portion of the heat being delivered to the spaces served.

133. Air Humidification with Warm-Air Furnaces. The water pan when placed in the casing, as shown by Fig. 53, is generally quite ineffective in evaporating the required amount of moisture for air humidification. Better results are obtainable with humidifiers located either in the bonnet of a furnace of the type shown by Fig. 53 or in the upper part of the casing of units similar to the one illustrated by Fig. 58. These placements are where the air temperatures are higher and where the air has a greater capacity for absorbing moisture.

The amount of moisture required per 24 hr in a structure is dependent on the desired relative humidity of the inside air, the amount of air leaking into and out of the building, and the temperature and relative humidity of the outside air. The moisture necessary for humidification can be calculated by the method of the following problem.

Example. A residence having 12,000 cu ft of occupied space is to be maintained at 71.5 dbt and 35 per cent relative humidity to secure comfort conditions as indicated by the comfort chart, Fig. 18. The house has one air change per hour when the outside-air temperature is 0 F, and it has a relative humidity of 100 per cent. Find the gallons of water required per 24 hr for humidification.

Solution. The weight of dry air leaking into and out of the house per hour at 71.5 F and atmospheric pressure is $12{,}000 \times 0.07471 = 896.5$ lb. Air at 71.5 F and 35 per cent rh (relative humidity), Fig. 5, contains 41 grains of moisture per pound of dry air. At 0 F and 100 per cent relative humidity the moisture of one pound of dry air is 5.5 grains. The moisture to be added per 24 hr is

$$[24 \times 896.5(41 - 5.5)] \div 7000 = 109.1 \text{ lb}$$

With water supplied at a density of 62.3 lb per cu ft the number of gallons is $(109.1 \times 1728) \div (231 \times 62.3) = 13.1$.

134. Furnace Sizes and Ratings. The size of a furnace, designed to burn solid fuels, is usually designated by the maximum diameter of the firepot taken at a location just below the feedneck section. Thus a 28-in. furnace has a maximum firepot diameter of that dimension at the specified point. The nominal diameter of the grate is that of the bottom of the firepot at a point just above the asphit section. The nominal grate diameter is usually less than the listed furnace size because of the taper of some cast-iron firepots and the refractory linings of steel furnaces.

Capacities of furnaces are commonly expressed in terms of Btu per hour delivered either at the furnace bonnet or at the warm-air registers. Extensive research in the field of warm-air heating, covering a period of many years, at the University of Illinois has resulted in the formulation of several codes by the National Warm Air Heating and Air Conditioning Association.

Gravity Ratings by Code. A code for the rating of furnaces used in gravity systems, adopted and recommended in December 1944, should supersede all prior ones. This last code, although based on factual data, is somewhat empirical and embodies certain definitions and limitations.

For gravity-flow warm-air furnaces with different methods of firing, capacity ratings by code are as given by Table 32.

TABLE 32

CAPACITY RATINGS OF GRAVITY-FLOW WARM-AIR FURNACES BY CODE

Furnace	Ratio of Heating Surface to Grate Area	Register Delivery, Btu per Hr	Bonnet Delivery, Btu per Hr
Hand-fired converted to stoker, oil or gas, firing		$1785 \times S$	$1.33 \times 1785 \times S$
Hand-fired	Greater than 15 to 1 and less than 25 to 1	$1785 \times S$	$1.33 \times 1785 \times S$
Hand-fired	Greater than 25 to 1	$1785 \times 25 \times G$	$1.33 \times 1785 \times 25 \times G$

The definitions are: (1) G is actual grate area, sq ft; (2) S equals actual heating surface, sq ft; (3) the register delivery is 75 per cent of the heat supplied to leaders at the bonnet; and (4) the bonnet capacity is 1.33 times the register delivery in Btu per hour.

For coal-burning furnaces, used in gravity-flow systems, the code limitations provide that the following shall not be in excess of: (1) register temperature, 175 F; (3) combustion rate, 8 lb per hr per sq ft of grate area; (3) flue-gas temperature, 800 F; (4) metal-surface temperature, at any point, 1000 F; and (5) draft loss from ashpit to smoke collar, 0.08 in. water gage (WG).

Actual heating surface S is measured above the grate level of a furnace and is the area with which circulated air actually comes in contact. This surface does not include that of liners, casings, radiation shields, etc. Extended surfaces such as fins, ribs, webs, and projections over which air to be warmed flows are taken as heating surfaces subject to limitations and restrictions. Their effectiveness decreases with the distance of their surfaces from the point of attachment to the furnace proper.

The actual grate area is smaller than the nominal grate area based on the firepot diameter at a location just above the ashpit. The actual grate area G excludes from the nominal grate area the area of an internal ledge of a firepot which falls within a circle one inch smaller in diameter than the firepot diameter inside its linings at the grate level. Also the areas of internal projections falling within a circle 3 in. smaller in diameter than that of the firepot at the grate level are deducted from the nominal grate area in computing the actual grate area. For inclined and conical grates the projected area of the grate is used in finding the actual area.

Rating formulas, for gravity systems, are based on a bonnet efficiency of 55 per cent for coal furnaces and 75 per cent for gas-fired units. Ratings on the basis of leader cross-sectional area, now used less frequently than formerly, allow 136 Btu per hr per sq in. as the average carrying capacity of a duct when the equivalent register temperature at the registers is 175 F with one-half the leaders serving first-floor rooms and the remainder of them supplying air to second-floor spaces. Equivalent register temperature is 65 F plus the actual register temperature minus the temperature of the air as it enters the furnace casing.

Warm-air furnaces designed for the use of either oil or gas alone do not include grates and are often quite different in their construction than those built for the use of coal alone. The differences in design lead to more efficient transfer of heat from the products of combustion to the circulating air in furnaces built for the exclusive use of either oil or gas, and also less consideration has to be given to the accessibility of their heating surfaces for cleaning operations. Special designs of furnaces for the burning of either oil or gas as a fuel are more efficient in operation than those built for the use of coal as a fuel and then later fitted with either conversion oil or gas burners as discussed in Chap. 5. Furnaces designed for the exclusive use of either oil or gas as a fuel are generally rated in terms of heat of the fuel input to the furnace, heat output at the furnace bonnet, or the heat delivered by the warm-air registers, all in Btu per hour.

Commercial Furnaces. Typical physical dimensions and ratings for one particular steel furnace are given by the data of Table 33.

TABLE 33

PHYSICAL DIMENSIONS AND RATINGS OF LENNOX ROUND CASED COAL-BURNING STEEL FURNACES

Firepot Diameter, In.	Casing Diameter, In.	Smokepipe Diameter, In.	Gravity Casing Height with 14-in. Bonnet, In.	Forced-Air Casing Height to Base of Plenum, In.	Ratings in Btu per Hour			
					Gravity		Forced-air	
					At Bonnet	At Register	At Bonnet	At Register
22	43	9	67½	59	114,070	85,550	127,700	108,550
24	45	9	67½	59	119,570	89,680	133,870	113,790
27	50	9	67½	59	146,350	109,760	163,850	139,270
30	54	10	72	64	181,450	136,090	203,150	172,680

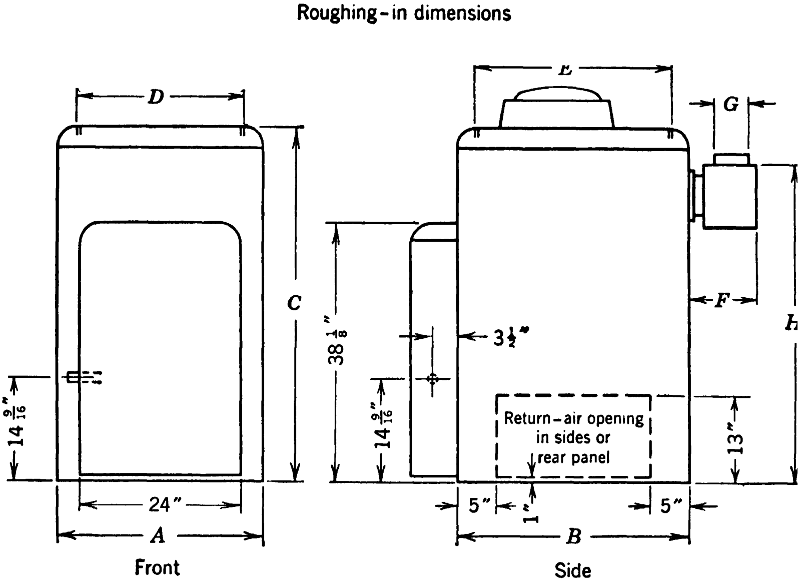
Figure 57 and Table 34 indicate typical dimensions and performance data for two sizes of gravity-flow warm-air furnaces built for the sole use of gaseous fuels.

TABLE 34

CAPACITY DATA FOR JANITROL GRAVITY-FLOW WARM-AIR FURNACES

Catalog Number	Rating, Btu per Hour		Gas Capacity, Cu Ft per Hr with Btu Content of			Pipe Area, Sq. In.		Approximate Shipping Weight, Lb
	Input AGA	Output at Register	550	800	1000	First-Floor Leaders	Second-Floor Leaders	
GCS 75-64	75,000	51,000	136	90	75	459	305	300
GCS100-64	100,000	68,000	182	125	100	613	407	350

Forced-Circulation Ratings by Code. Mechanical warm-air furnaces are rated in terms of Btu per hour, based on either register delivery or bonnet capacity. The register-delivery capacity with mechanical circulation is assumed as 85 per cent of the output at the bonnet. If practicable, ratings should be determined by actual test conducted under the conditions which have been specified for the testing of gravity-flow furnaces. Furnaces designed for the burning of solid fuels and for use in forced-circulation systems for which rating tests have not been made may be rated from the data of Table 35.



Dimensions in Inches

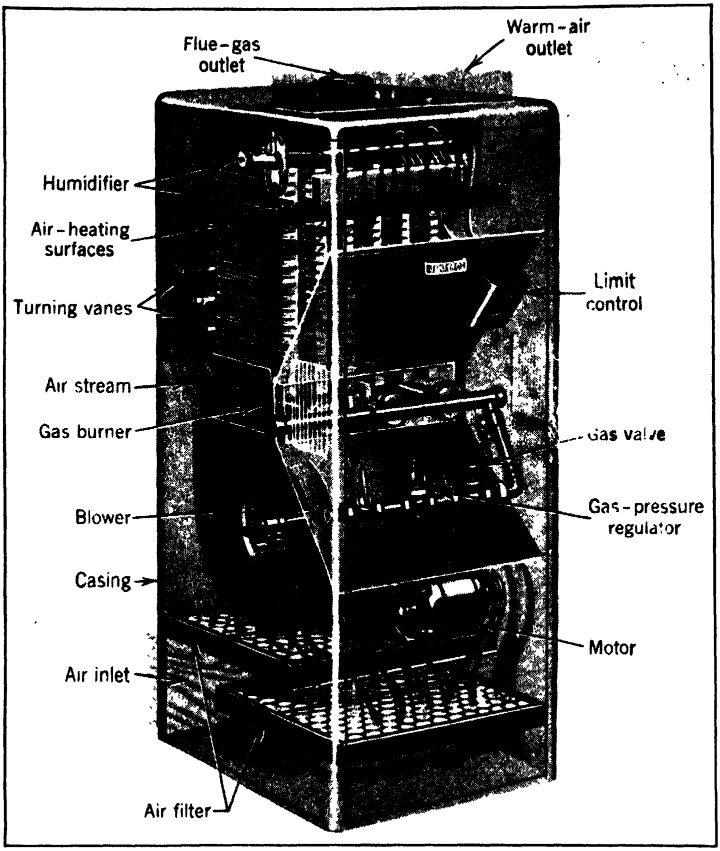
Catalog Number	A	B	C	D	E	F	G	H	Gas Connection*	
									Natural Mixture	Manufactured
GCS 75-64	30	34	$52\frac{1}{4}$	$27\frac{9}{16}$	$31\frac{9}{16}$	10	5	$49\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
GCS100-64	36	38	$52\frac{1}{4}$	$33\frac{9}{16}$	$35\frac{9}{16}$	11	6	50	$\frac{3}{4}$	$\frac{3}{4}$

* This is the manifold connection, not the recommended size of the gas supply line.

(*Surface Combustion Corp.*)

Fig. 57. Dimensions of Janitrol gas-fired gravity-flow warm-air furnaces.

Many manufacturers offer oil-burning and gas-burning packaged units which are designed specifically for forced circulation. Figure 58 shows the interior view of such a unit designed for the use of gas as



(*Surface Combustion Corp.*)

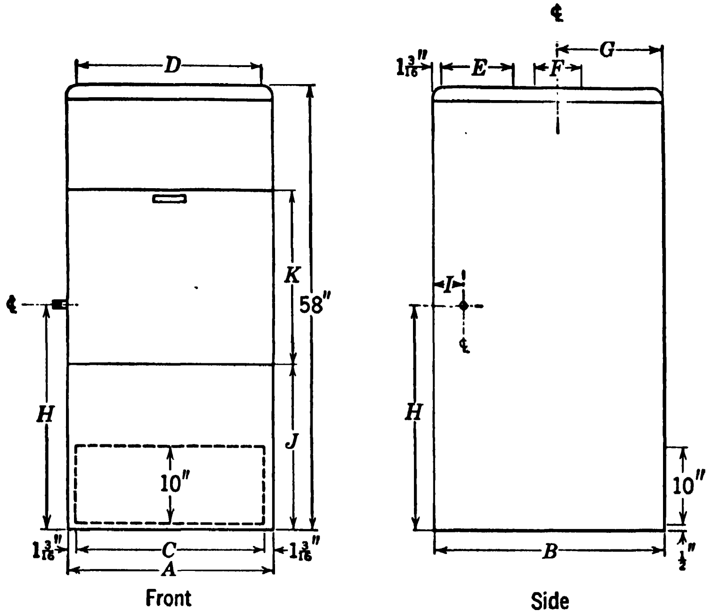
FIG. 58. Typical gas-fired packaged-unit forced-circulation warm-air furnace.

TABLE 35

CAPACITY RATINGS OF FORCED-CIRCULATION WARM-AIR FURNACES BY CODE

Furnace	Ratio of Heating Surface to Grate Area	Register Delivery, Btu per Hr	Bonnet Delivery, Btu per Hr
Hand-fired converted to stoker, oil, or gas firing		2265 × S*	1.18 × 2265 × S
Hand-fired	Greater than 15 to 1 and less than 25 to 1	2265 × S*	1.18 × 2265 × S
Hand-fired	Greater than 25 to 1	2265 × 25G*	1.18 × 2265 × 25 × G

* S = actual heating surface, sq ft. G = actual grate area, sq ft.



DIMENSIONS IN INCHES

Catalog Number	A	B	Return Air		Warm Air		Flue‡ F	G	H	I	J	K	Gas Connection*		Filter		Motor†	
			Ht	Width C	D	E							Natural	Manufactured	Number	Size	HP	Type
FAC 60-14	$18\frac{3}{8}$	27	10	16	16	8	5	$11\frac{3}{4}$	$26\frac{1}{4}$	$2\frac{5}{8}$	20	$28\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	2	16 × 20	$\frac{1}{6}$	SP
FAC 90-14	$22\frac{3}{8}$	27	10	20	20	8	5	$11\frac{3}{4}$	$26\frac{1}{4}$	$2\frac{5}{8}$	20	$28\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	2	20 × 20	$\frac{1}{4}$	SP
FAC 105-14	$24\frac{3}{8}$	27	10	22	22	8	6	$11\frac{3}{4}$	$26\frac{1}{4}$	$2\frac{5}{8}$	20	$28\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	2	20 × 20	$\frac{1}{4}$	SP
FAC 120-14	$27\frac{3}{8}$	30	10	25	25	10	6	$12\frac{3}{4}$	29	$3\frac{7}{8}$	$21\frac{7}{8}$	$25\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	2	25 × 20	$\frac{1}{4}$	SP
FAC 150-14	$34\frac{3}{8}$	30	10	32	32	10	7	$12\frac{3}{4}$	29	$3\frac{7}{8}$	$21\frac{7}{8}$	$25\frac{1}{4}$	$\frac{3}{4}$	1	4	16 × 20	$\frac{1}{3}$	SP
FAC 180-14	$42\frac{3}{8}$	30	10	40	40	10	7	$12\frac{3}{4}$	29	$3\frac{7}{8}$	$21\frac{7}{8}$	$25\frac{1}{4}$	1	1	4	20 × 20	$\frac{1}{3}$	SP

* This is size of manifold connection; not recommended size of the gas supply line. SP—Split phase.

† Standard motors are 1725 rpm for 115-volt, 60-cycle, single-phase current.

‡ This is the draft diverter outlet dimension.

(*Surface Combustion Corp.*)

FIG. 59. Physical dimensions of Janitrol packaged-type gas-fired forced-circulation warm-air furnaces.

a fuel. Included within a single casing are filters, a fan, burners, and the required amount of heat-transfer surface. Physical dimensions are shown by Fig. 59 and rating data are given for several different sized furnaces of this type in Table 36.

TABLE 36

RATING DATA FOR PACKAGED-TYPE, GAS-FIRED, FORCED-CIRCULATION, WARM-AIR FURNACES*

Model Number	Rating Btu, per Hr		Cfm Delivery for Stated, Deg F, Temperature Rise			Gas Cap., Cu Ft Hr with Btu Content of			Blower-Wheel Data			Rpm at 0.15 In. of Water and 80 F	Approximate Shipping Weight, Lb
	Input	Output at Register	70°	80°	90°	550	800	1000	Diameter	Width	Number		
FAC 60-14	60,000	43,200	600	525	465	109	75	60	13	7	1	510	400
FAC 90-14	90,000	64,800	900	790	700	164	113	90	13	10	1	535	480
FAC 105-14	105,000	75,600	1050	920	815	191	131	105	13	10	1	580	600
FAC 120-14	120,000	86,400	1200	1050	935	218	150	120	13	12	1	575	650
FAC 150-14	150,000	108,000	1500	1310	1170	273	188	150	13	7	2	510	780
FAC 180-14	180,000	129,600	1800	1575	1400	328	225	180	13	10	2	520	904

* Surface Combustion Corporation.

135. Casing Bonnets for Gravity Flow. The height of the bonnet of a gravity-flow furnace must be great enough so that the largest leader used can be attached to it. Collars for the leaders are necessary at the outlet openings. Bonnets, Fig. 60, are constructed with pitched and straight sides. When the leaders are taken from the bonnet sides the top of the bonnet is an inverted cone. Where the leaders are taken from the bonnet top, the ducts are attached to a flat surface. The sloped bonnets are of aid in giving the leaders pitch. Where the leaders are attached to straight side bonnets a partial elbow at the collar is necessary. Bonnets with vertical sides, as *C* of Fig. 60, have the outlets in the top and require an elbow at the fastenings of the leaders to the bonnet to give them the proper direction and pitch. This latter arrangement requires more basement headroom than when the leader attachments are at the sides. According to the results given in *Bulletin* 141 of the University of Illinois Engineering Experiment Station,[1] type B is the best for all conditions of service.

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[1] "Investigations of Warm-Air Furnaces and Heating Systems," by A. C. Willard, A. P. Kratz, and V. S. Day.

136. Leader Pipes for Gravity-Flow Systems. Commercial sizes of round ducts range as follows: 8-, 9-, 10-, 12-, and 14-in. diameter. These pipes are generally made of light sheet steel plated with tin or of sheet aluminum; they may also be galvanized steel. Leaders less than 8 in. in diameter are not practical in gravity-flow installations because of excessive friction and heat losses. Therefore, irrespective of the calculated size, no gravity-flow leader should be installed which has a diameter less than 8 in. For gravity-flow installations leaders larger than 14 in. in diameter involve the use of bonnets higher than those generally provided with furnace casings.

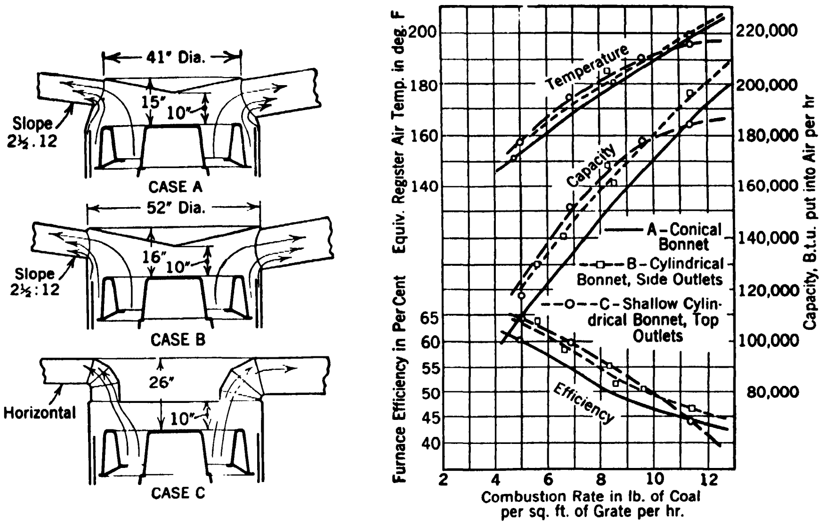


FIG. 60. Performance curves for three types of gravity-flow warm-air furnace bonnets.

Leaders not exceeding 6 to 8 ft in length are the most desirable, although leaders 10 to 12 ft long can be used. Wherever a gravity-flow leader exceeds 12 ft in length special consideration must be given to it. If possible, all leaders should be of approximately the same length. In order to control the air flow in the leaders of varying length it is necessary to have a cross damper installed in each leader at a location near to the furnace bonnet.

The heat-carrying capacity of a leader is dependent upon its length, the resistance of the included fittings, the temperature of the air carried by the leader, the inside-air temperature of the structure, and the height of the register outlet above the furnace. Thus it follows that first-floor leaders will have a smaller heat-carrying capacity per unit of cross-sectional area, under fixed operating conditions, than leaders serving either second-floor or third-floor rooms. Likewise the heating capacity of a leader of a given diameter serving a second-floor room is less than that of a leader of equal diameter serving a third-floor room under the same temperature conditions. For gravity-flow warm-air-furnace plants operating with 70 F at the room breathing-line levels and 175 F equivalent-register temperatures, Art. 134, commonly used values of leader heat-carrying capacities have been: first-floor, 111; second-floor, 167; and third-floor rooms, 200 Btu per hr per sq in. of leader cross-sectional area. The data and methods which are given in Art. 140 permit more accurate and satisfactory selections of leader pipes, fittings, stacks, and registers for one- and two-story structures than those methods based on Btu per hour per square inch of leader cross section.

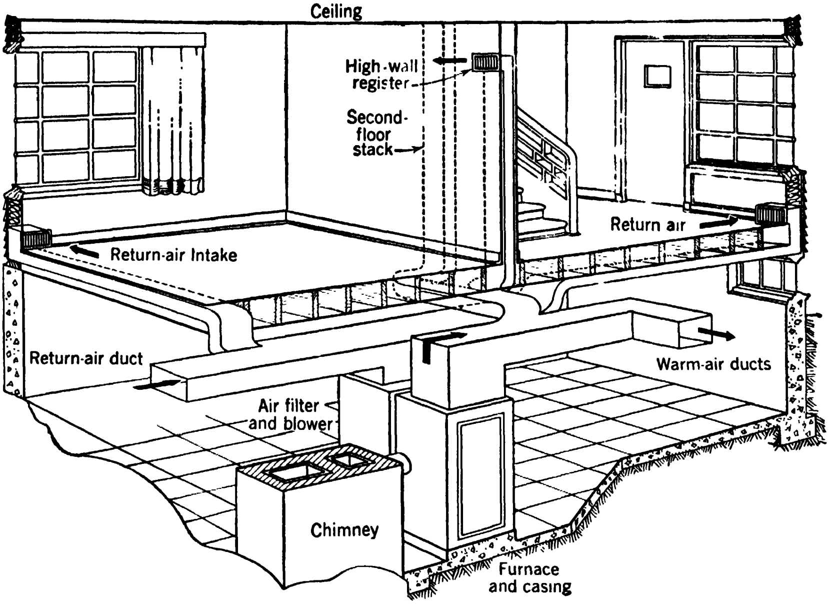


FIG. 61. Diagrammatic section of a typical forced-circulation warm-air heating system.

137. Duct Arrangements for Mechanical Systems. The gravity-flow system of Fig. 52 could be converted to forced circulation of the air by the addition of a fan and a revision of the recirculating duct, transition piece, and shoe. However, systems initially designed for forced circulation usually include a different type of bonnet, and both warm-air and return-air ducts are arranged to interfere as little as possible with the use of the floor area beneath them. Figure 61 shows a typical piping arrangement for conducting the heated air away from the plenum chamber at the top of the furnace. The illustration also shows part of the ducts required to collect the return air and conduct it to the lower part of the furnace casing. In some mechanical systems the fan and the filter are placed in the lower portion of the furnace casing (see Fig. 58), while in other arrangements, as in Fig. 61, these parts of the system are housed in a separate cabinet which is adjacent to the furnace casing. Regardless of the location of the fan and filter it is the usual practice to design the return-air trunk in such a way that a minimum amount of floor area is made useless by its presence.

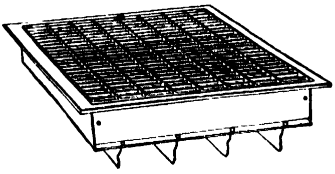
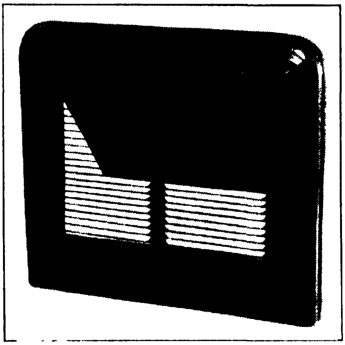


Fig. 62. Floor register.

138. Registers in Gravity-Flow Systems. A grille with a control valve of some type should be placed over or in openings through which the warm air enters spaces, and the entrances to all recirculating ducts should be protected in a suitable manner by valveless units. Warm-air registers in gravity-flow plants may be located in the floor adjacent to an interior partition, in an interior partition at the level of the baseboard, or in an interior partition just above the level of the baseboard. Figure 62 shows a typical floor register with a control valve consisting of four movable vanes. A modern baseboard type of register is shown in Fig. 63. The small handle appearing in the upper center of the photograph is for the operation of the control valve located at the rear of the grille.

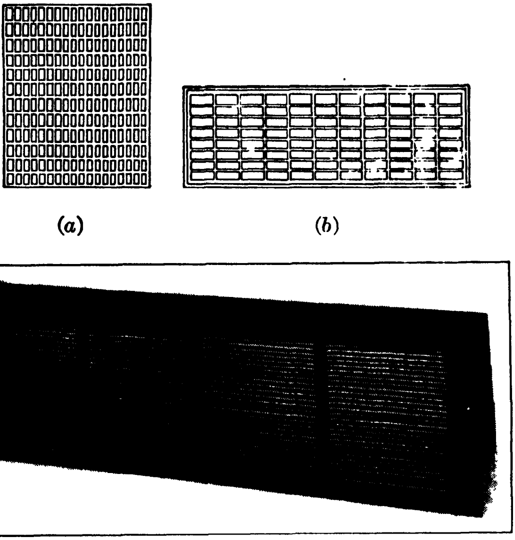


(Turnbull Manufacturing and Distributing Co.)

Fig. 63. Baseboard register for a gravity-flow system.

In general, one register is provided for each room, though large living rooms may require two and small rooms may be heated from a register in an adjacent room provided that the two rooms are always interconnected by an archway that is at least 7 ft high.

It is usual practice to locate all return-air registers in gravity-flow plants on the first floor of the house. If the house includes a second floor, one register located near the base of the stairway returns all the recirculated air from the second floor. This register which will invariably be larger than any of the others may also return part of the air from the first-floor rooms. If practicable, one or more additional return-air registers should be located near outside walls where they will be most effective in intercepting down drafts of cold air from glass surfaces, thus preventing the movement of cold air across the floor of the living room. Doors of rooms such as bedrooms which are not provided with return-air outlets must clear the floor or floor covering by at least $\frac{1}{2}$ in. to permit the flow of return air to the nearest register. Three different types of return-air registers for gravity-flow systems are illustrated in Fig. 64.


(a) (b)

(c)

FIG. 64. Return-duct registers for gravity flow. (a) Wood. (b) Metal for floor use. (c) Metal baseboard.

139. Registers in Mechanical Warm-Air Systems. When air circulation is produced by means of a fan, satisfactory heating of a room may be produced with the warm-air registers located in the floor, low on a side wall, high on a side wall, or in the ceiling. The principal requirement from a comfort standpoint is that the register is so located that the air stream does not strike an occupant before it has lost its high velocity. The high sidewall location is extensively used because when the register is in this position there is little chance that the air stream will strike an occupant anywhere in the room. A register located high on a side wall offers an additional advantage in that it does not interfere with any furniture arrangement which may be desired. The ceiling register also offers these advantages but is less commonly used because when the furnace is located in the basement, more piping is required to reach a warm-air outlet in this location. A register suitable for either the high sidewall or low sidewall location in a forced-circulation system is shown in Fig. 65*b*. Other types of outlet grilles suitable for use in forced-circulation systems are discussed in Chap. 13.

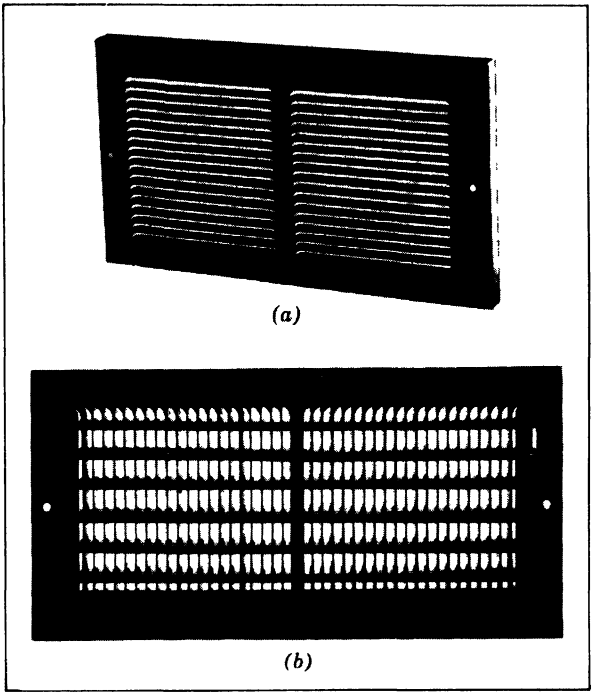


FIG. 65. Forced-circulation warm-air registers. (*a*) Baseboard return-air unit. (*b*) High- or low-sidewall register arranged to deflect one-half of air to right and one-half to left.

In designing forced-circulation warm-air systems it is the usual practice to include a return-air grille in each room to be heated with the exception of bathrooms, closets, kitchens, and lavatories. A vent to the outside should be provided in the kitchen to remove cooking odors from the house. Large living rooms are usually provided with two return-air intakes. Intake grilles for mechanical systems are generally placed in an outer wall of the room with the lower edge at floor level. They have the general appearance of the sidewall outlet registers but do not include a means of regulating the flow of air. A typical return-air grille, for forced circulation, is shown in Fig. 65*a*. Floor intakes may also be used and offer less resistance to the flow of air than those placed in a vertical position. However, the sidewall type of intake is usually preferred because it does not interfere with the placement of any type of floor covering, and there is less chance that small articles will accidentially drop into it.

140. Design of Piped Gravity-Flow Warm-Air Heating Plants. The Installation Codes Committee of the National Warm Air Heating and Air Conditioning Association recommends the following procedure[2] for determining the proper sizes and specifications for the component parts of gravity-flow warm-air heating systems suitable for one- and two-story residences. The various details of design are:

1. Locate all warm-air registers and return-air intakes on the first- and second-floor plans of the house to be heated; indicate floor- and baseboard-register placements as shown by Fig. 66. Accepted symbols and abbreviations for such plans are detailed by Fig. 67.

2. Indicate any required crossover connections between register and riser or stack locations. A crossover connection is a horizontal rectangular duct placed between floor joists as indicated by offsets *g*, *h*, and *i* of Fig. 68.

3. On a basement plan, similar to the one of Fig. 66, show the locations of the following items:
   a. Warm-air and return risers for first- and second-floor rooms.
   b. Furnace, including front of furnace.
   c. Smokepipe or flue connections to chimney.
   d. Warm-air leader pipes to risers or stacks.
   e. Return-air branch lines.
   f. Return-air ducts, to furnace, serving one or more branch lines.

4. Calculate the heat losses from each room according to the methods outlined in Chap. 4 or with the aid of a suitable table.

5. Determine the actual length, in feet, of each warm-air leader from the layout on the basement plan. Do not include the length of vertical risers; the horizontal length of any crossover connection should be added to the horizontal length of its leader as determined from the basement plan.

6. Determine the number of right-angle elbows which would offer the same resistance to air flow as would the fittings in any leader or leader and crossover combination. Do not consider the registers in

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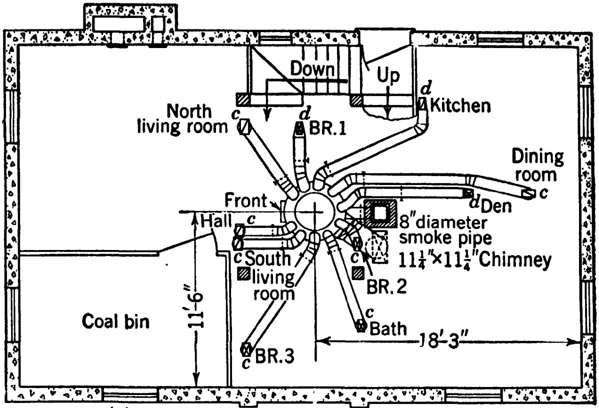
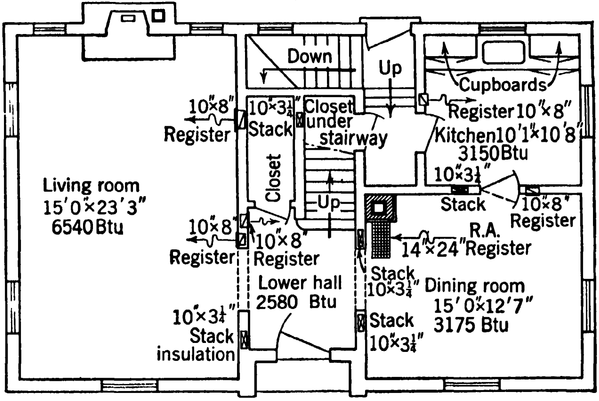
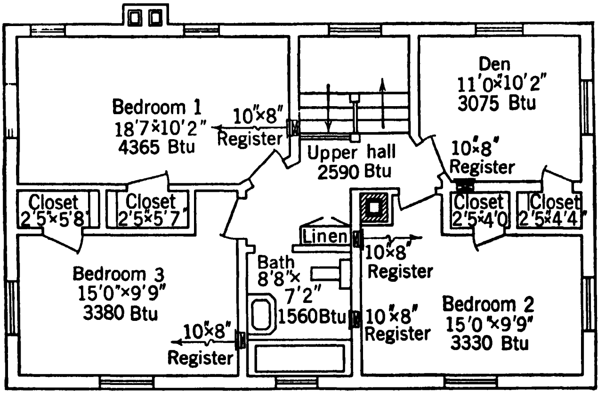
[2] *"Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems,"* published by the NWAH and ACA of Cleveland, Ohio.



(*a*) Plan of basement, all leaders 8-in. diameter

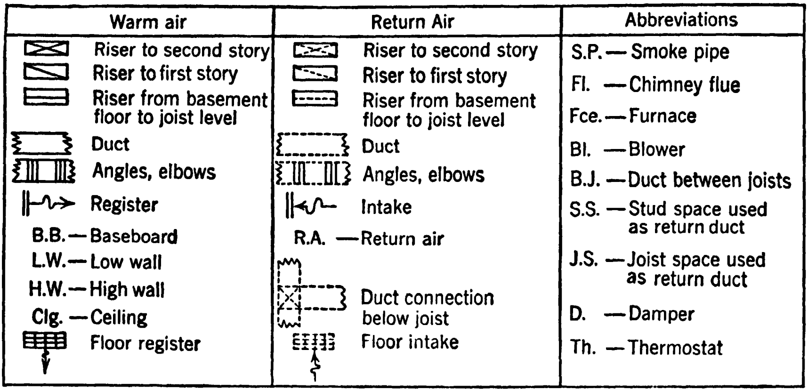


(*b*) First floor



(*c*) Second floor

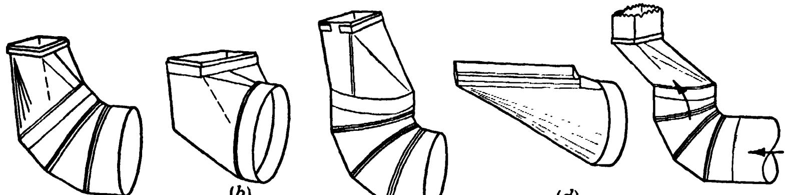
FIG. 66. Layout of a gravity-flow warm-air furnace system in a residence.



Warm air	Return Air	Abbreviations
Riser to second story	Riser to second story	S.P.—Smoke pipe
Riser to first story	Riser to first story	Fl. —Chimney flue
Riser from basement floor to joist level	Riser from basement floor to joist level	Fce.—Furnace
Duct	Duct	Bl. —Blower
Angles, elbows	Angles, elbows	B.J. —Duct between joists
Register	Intake	S.S. —Stud space used as return duct
B.B.—Baseboard	R.A. —Return air	J.S. —Joist space used as return duct
L.W.—Low wall	Duct connection below joist	D. —Damper
H.W.—High wall	Floor intake	Th. —Thermostat
Clg. —Ceiling		
Floor register		

FIG. 67. Symbols and abbreviations for warm-air heating plans.

making allowances for fittings. Elbow equivalents of commonly used fittings are given by Fig. 68. The boots shown in Fig. 68 are available in two different styles. One style has top outlet openings equal to the proper throat dimensions listed in Section *A* of Table 39, and they are used when serving baseboard registers located in a first-



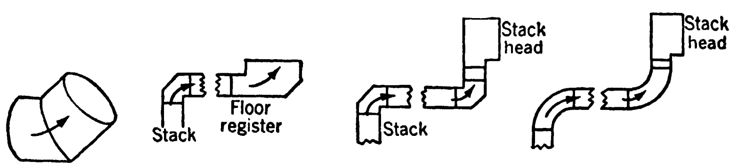
(*a*) 45-deg angle boot and 45-deg elbow. Consider combination as equivalent to *one elbow*

(*b*) 90-deg angle boot. Consider combination as equivalent to *one elbow*

(*c*) Universal boot and 90-deg elbow. Consider combination as equivalent to *one elbow*

(*d*) End boot. Consider combination as equivalent to *two elbows*

(*e*) Offset boot. Consider combination as equivalent to *2½ elbows*



(*f*) 45-deg angle. Consider as equivalent to *½ elbow*

(*g*) Floor register second story. Consider combination as equivalent to *three elbows*

(*h*) Offset. Consider combination as equivalent to *three elbows*

(*i*) Offset. Consider combination as equivalent to *two elbows*

FIG. 68. Warm-air boot combinations and their elbow equivalents.

floor room. The other style has top outlet openings equal to the proper stack dimensions shown in Section *B* of Table 39 and they are used when serving any type of register located on the second floor.

TABLE 37

Warm-Air Carrying Capacity, First-Story Registers for Gravity Flow

Warm-Air Combination Number	Actual Length Of Leader, from Bonnet to Boot, Ft											Warm-Air Combination Number
	4 Column *a*	6 Column *b*	8 Column *c*	10 Column *d*	12 Column *e*	14 Column *f*	16 Column *g*	18 Column *h*	20 Column *i*	22 Column *j*	24 Column *k*	
	Section *A*, Btu per Hr Delivery with One Elbow											
1	6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320	1
2	7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460	2
3	9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740	3
4	13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560	4
5	17,520	17,020	16,530	16,040	15,550	15,050	14,550	14,050	13,560	13,060	12,560	5
	Section *B*, Btu per Hr Delivery with Two Elbows											
1	5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180	1
2	7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290	2
3	9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520	3
4	12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210	4
5	16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150	5
	Section *C*, Btu per Hr Delivery with Three Elbows											
1	5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030	1
2	7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100	2
3	8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300	3
4	12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950	4
5	16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750	5
	Section *D*, Btu per Hr Delivery with Four Elbows											
1	5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890	1
2	6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910	2
3	8,460	8,220	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070	3
4	12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,290	8,950	8,610	4
5	15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310	5
	Section *E*, Btu per Hr Delivery with Five Elbows											
1	5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760	1
2	6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750	2
3	8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860	3
4	11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320	4
5	15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940	5
	Section *F*, Btu per Hr Delivery with Six Elbows											
1	4,990	4,850	4,710	4,570	4,430	4,290	4,150	4,010	3,860	3,720	3,580	1
2	6,320	6,140	5,960	5,780	5,610	5,430	5,240	5,070	4,890	4,710	4,530	2
3	7,800	7,580	7,360	7,140	6,920	6,700	6,480	6,260	6,030	5,820	5,590	3
4	11,070	10,760	10,450	10,140	9,820	9,500	9,200	8,880	8,560	8,260	7,930	4
5	14,540	14,120	13,710	13,310	12,900	12,500	12,070	11,650	11,250	10,840	10,430	5
	Section *G*, Btu per Hr Delivery with Seven Elbows											
1	4,820	4,680	4,540	4,410	4,270	4,140	4,000	3,860	3,730	3,590	3,460	1
2	6,100	5,920	5,740	5,580	5,410	5,230	5,060	4,890	4,710	4,540	4,370	2
3	7,520	7,310	7,100	6,880	6,670	6,460	6,250	6,030	5,820	5,610	5,390	3
4	10,680	10,370	10,070	9,770	9,460	9,160	8,860	8,560	8,260	7,960	7,650	4
5	14,000	13,610	13,220	12,830	12,440	12,040	11,640	11,240	10,850	10,450	10,050	5

When a leader serves a floor register in a room in the first story a special boot terminating in the dimensions of the register pan is used.

7. Use the information given by steps 4, 5, and 6 together with either Table 37 or 38 to find the numbers of the correct combinations to be used for supplying heated air to each room.

TABLE 38

Warm-Air Carrying Capacity, Second-Story Registers for Gravity Flow

<table>
<tr><th rowspan="2">Warm-Air Combination Number</th><th colspan="11">Actual Length of Leader, From Bonnet To Boot, Feet</th><th rowspan="2">Warm-Air Combination Number</th></tr>
<tr><th>4 Column <i>a</i></th><th>6 Column <i>b</i></th><th>8 Column <i>c</i></th><th>10 Column <i>d</i></th><th>12 Column <i>e</i></th><th>14 Column <i>f</i></th><th>16 Column <i>g</i></th><th>18 Column <i>h</i></th><th>20 Column <i>i</i></th><th>22 Column <i>j</i></th><th>24 Column <i>k</i></th></tr>
<tr><td></td><td colspan="11">Section <i>A</i>, Btu per Hr Delivery with One Elbow</td><td></td></tr>
<tr><td>11</td><td>8,370</td><td>8,140</td><td>7,900</td><td>7,670</td><td>7,430</td><td>7,190</td><td>6,950</td><td>6,710</td><td>6,470</td><td>6,240</td><td>6,000</td><td>11</td></tr>
<tr><td>12</td><td>10,040</td><td>9,760</td><td>9,470</td><td>9,190</td><td>8,900</td><td>8,620</td><td>8,330</td><td>8,050</td><td>7,770</td><td>7,480</td><td>7,200</td><td>12</td></tr>
<tr><td>14</td><td>11,710</td><td>11,380</td><td>11,050</td><td>10,720</td><td>10,390</td><td>10,060</td><td>9,720</td><td>9,390</td><td>9,060</td><td>8,730</td><td>8,400</td><td>14</td></tr>
<tr><td>15</td><td>16,200</td><td>15,750</td><td>15,300</td><td>14,840</td><td>14,380</td><td>13,920</td><td>13,460</td><td>13,000</td><td>12,550</td><td>12,100</td><td>11,640</td><td>15</td></tr>
<tr><td>16</td><td>18,920</td><td>18,390</td><td>17,850</td><td>17,310</td><td>16,780</td><td>16,240</td><td>15,710</td><td>15,180</td><td>14,640</td><td>14,100</td><td>13,570</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>B</i>, Btu per Hr Delivery with Two Elbows</td><td></td></tr>
<tr><td>11</td><td>7,940</td><td>7,720</td><td>7,500</td><td>7,280</td><td>7,050</td><td>6,830</td><td>6,600</td><td>6,370</td><td>6,150</td><td>5,930</td><td>5,700</td><td>11</td></tr>
<tr><td>12</td><td>9,540</td><td>9,270</td><td>9,000</td><td>8,730</td><td>8,460</td><td>8,190</td><td>7,920</td><td>7,650</td><td>7,380</td><td>7,110</td><td>6,840</td><td>12</td></tr>
<tr><td>14</td><td>11,120</td><td>10,810</td><td>10,500</td><td>10,180</td><td>9,870</td><td>9,550</td><td>9,230</td><td>8,920</td><td>8,610</td><td>8,290</td><td>7,980</td><td>14</td></tr>
<tr><td>15</td><td>15,400</td><td>14,970</td><td>14,530</td><td>14,100</td><td>13,670</td><td>13,230</td><td>12,800</td><td>12,360</td><td>11,930</td><td>11,500</td><td>11,070</td><td>15</td></tr>
<tr><td>16</td><td>17,980</td><td>17,470</td><td>16,960</td><td>16,450</td><td>15,950</td><td>15,430</td><td>14,830</td><td>14,420</td><td>13,910</td><td>13,400</td><td>12,890</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>C</i>, Btu per Hr Delivery with Three Elbows</td><td></td></tr>
<tr><td>11</td><td>7,530</td><td>7,320</td><td>7,110</td><td>6,900</td><td>6,680</td><td>6,470</td><td>6,250</td><td>6,040</td><td>5,830</td><td>5,620</td><td>5,400</td><td>11</td></tr>
<tr><td>12</td><td>9,030</td><td>8,780</td><td>8,520</td><td>8,270</td><td>8,010</td><td>7,750</td><td>7,500</td><td>7,240</td><td>6,990</td><td>6,730</td><td>6,470</td><td>12</td></tr>
<tr><td>14</td><td>10,530</td><td>10,240</td><td>9,940</td><td>9,650</td><td>9,350</td><td>9,050</td><td>8,750</td><td>8,450</td><td>8,160</td><td>7,860</td><td>7,560</td><td>14</td></tr>
<tr><td>15</td><td>14,580</td><td>14,180</td><td>13,780</td><td>13,370</td><td>12,950</td><td>12,530</td><td>12,120</td><td>11,710</td><td>11,300</td><td>10,890</td><td>10,480</td><td>15</td></tr>
<tr><td>16</td><td>17,040</td><td>16,550</td><td>16,070</td><td>15,580</td><td>15,110</td><td>14,620</td><td>14,140</td><td>13,660</td><td>13,180</td><td>12,700</td><td>12,210</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>D</i>, Btu per Hr Delivery with Four Elbows</td><td></td></tr>
<tr><td>11</td><td>7,120</td><td>6,920</td><td>6,720</td><td>6,520</td><td>6,310</td><td>6,110</td><td>5,900</td><td>5,700</td><td>5,500</td><td>5,300</td><td>5,100</td><td>11</td></tr>
<tr><td>12</td><td>8,530</td><td>8,290</td><td>8,050</td><td>7,810</td><td>7,570</td><td>7,330</td><td>7,080</td><td>6,840</td><td>6,600</td><td>6,360</td><td>6,120</td><td>12</td></tr>
<tr><td>14</td><td>9,950</td><td>9,670</td><td>9,390</td><td>9,110</td><td>8,830</td><td>8,550</td><td>8,260</td><td>7,980</td><td>7,700</td><td>7,420</td><td>7,140</td><td>14</td></tr>
<tr><td>15</td><td>13,780</td><td>13,390</td><td>13,000</td><td>12,610</td><td>12,220</td><td>11,830</td><td>11,440</td><td>11,050</td><td>10,670</td><td>10,280</td><td>9,890</td><td>15</td></tr>
<tr><td>16</td><td>16,080</td><td>15,620</td><td>15,170</td><td>14,710</td><td>14,260</td><td>13,810</td><td>13,350</td><td>12,900</td><td>12,440</td><td>11,980</td><td>11,530</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>E</i>, Btu per Hr Delivery with Five Elbows</td><td></td></tr>
<tr><td>11</td><td>6,700</td><td>6,510</td><td>6,320</td><td>6,130</td><td>5,940</td><td>5,750</td><td>5,560</td><td>5,370</td><td>5,180</td><td>4,990</td><td>4,800</td><td>11</td></tr>
<tr><td>12</td><td>8,040</td><td>7,810</td><td>7,580</td><td>7,350</td><td>7,130</td><td>6,900</td><td>6,670</td><td>6,440</td><td>6,220</td><td>5,990</td><td>5,760</td><td>12</td></tr>
<tr><td>14</td><td>9,370</td><td>9,110</td><td>8,850</td><td>8,580</td><td>8,310</td><td>8,050</td><td>7,780</td><td>7,510</td><td>7,250</td><td>6,980</td><td>6,720</td><td>14</td></tr>
<tr><td>15</td><td>12,970</td><td>12,600</td><td>12,240</td><td>11,870</td><td>11,500</td><td>11,140</td><td>10,770</td><td>10,400</td><td>10,040</td><td>9,680</td><td>9,310</td><td>15</td></tr>
<tr><td>16</td><td>15,140</td><td>14,710</td><td>14,280</td><td>13,850</td><td>13,420</td><td>13,000</td><td>12,570</td><td>12,140</td><td>11,710</td><td>11,280</td><td>10,850</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>F</i>, Btu per Hr Delivery with Six Elbows</td><td></td></tr>
<tr><td>11</td><td>6,280</td><td>6,100</td><td>5,920</td><td>5,770</td><td>5,570</td><td>5,390</td><td>5,210</td><td>5,030</td><td>4,850</td><td>4,680</td><td>4,500</td><td>11</td></tr>
<tr><td>12</td><td>7,530</td><td>7,320</td><td>7,100</td><td>6,890</td><td>6,670</td><td>6,460</td><td>6,250</td><td>6,040</td><td>5,830</td><td>5,610</td><td>5,400</td><td>12</td></tr>
<tr><td>14</td><td>8,780</td><td>8,540</td><td>8,290</td><td>8,040</td><td>7,800</td><td>7,550</td><td>7,290</td><td>7,040</td><td>6,800</td><td>6,550</td><td>6,300</td><td>14</td></tr>
<tr><td>15</td><td>12,150</td><td>11,800</td><td>11,470</td><td>11,140</td><td>10,780</td><td>10,440</td><td>10,100</td><td>9,750</td><td>9,420</td><td>9,080</td><td>8,730</td><td>15</td></tr>
<tr><td>16</td><td>14,180</td><td>13,780</td><td>13,380</td><td>12,980</td><td>12,580</td><td>12,180</td><td>11,780</td><td>11,380</td><td>10,980</td><td>10,580</td><td>10,180</td><td>16</td></tr>
<tr><td></td><td colspan="11">Section <i>G</i>, Btu per Hr Delivery with Seven Elbows</td><td></td></tr>
<tr><td>11</td><td>5,860</td><td>5,700</td><td>5,530</td><td>5,370</td><td>5,200</td><td>5,040</td><td>4,870</td><td>4,700</td><td>4,530</td><td>4,370</td><td>4,200</td><td>11</td></tr>
<tr><td>12</td><td>7,020</td><td>6,830</td><td>6,630</td><td>6,440</td><td>6,230</td><td>6,040</td><td>5,830</td><td>5,640</td><td>5,440</td><td>5,240</td><td>5,040</td><td>12</td></tr>
<tr><td>14</td><td>8,200</td><td>7,970</td><td>7,740</td><td>7,500</td><td>7,280</td><td>7,040</td><td>6,800</td><td>6,570</td><td>6,340</td><td>6,110</td><td>5,880</td><td>14</td></tr>
<tr><td>15</td><td>11,340</td><td>11,020</td><td>10,710</td><td>10,380</td><td>10,070</td><td>9,740</td><td>9,420</td><td>9,100</td><td>8,790</td><td>8,470</td><td>8,150</td><td>15</td></tr>
<tr><td>16</td><td>13,240</td><td>12,870</td><td>12,500</td><td>12,120</td><td>11,750</td><td>11,370</td><td>11,000</td><td>10,630</td><td>10,250</td><td>9,870</td><td>9,500</td><td>16</td></tr>
</table>

When floor registers are used in second-story rooms, include the equivalent elbow resistance for the crossover connection, as shown in Fig. 68(*g*).

8. Use Table 39 to translate the correct combination number obtained from either Table 37 or 38 into actual dimensions of leaders, stacks, and registers.

The combinations of leader diameters and stack or throat sizes shown in Table 39 have been found to function satisfactorily, and an effort is being made to secure industry-wide standardization on these proportions. Dimensions other than those shown by Table 39 can

TABLE 39

Gravity-Flow Warm-Air Duct System, Combination of Parts Selected as Standard

Section *A*

First-Story Warm-Air Ducts

Warm-Air Combination Number	Leader-Pipe Diameter, In.	Pipe Area, Sq In.	Register Size, In.			
			a Floor	*b* Baseboard		
				Size	Extension	Throat Size
1	8	50	8 × 10	10 × 8	2¼	6⅛ × 10
2	9	64	9 × 12	12 × 8	2¼	6⅛ × 12
3	10	78	10 × 12	12 × 9	3¼	7⅛ × 12
4	12	113	12 × 14	13 × 11	5¼	9⅛ × 13
5	14	154	14 × 16	...	...	...

Note. When the calculations indicate a requirement for a given room greater than unit number 4, two or more smaller units totalling the required capacity are recommended.

Section *B*

Second-Story Warm-Air Ducts, Single-Wall Stacks and Fittings

Warm-Air Combination Numbers	Leader-Pipe Diameter, In.	Pipe Area, Sq In.	Stack* Size, In.	Register Size, In.			
				a Floor	*b* Baseboard		*c* Sidewall
					Size	Extension	
11	8	50	10 × 3¼	8 × 10	10 × 8	2¼	10 × 8
12	9	64	12 × 3¼	9 × 12	12 × 8	2¼	12 × 8
14	10	78	14 × 3¼	...	12 × 8	2¼	12 × 8
15	12	113	12 × 5¼	...	12 × 9	3¼	...
16	12	113	14 × 5¼	...	13 × 11	5¼	...

* Recommended stack sizes. This table may also be applied to 3- and 3½-in. stack depths.

be used, but the cross-sectional area of the stack should be at least 70 per cent of that of the leader attached to it.

9. Calculate the sum of the heat losses, in Btu per hour, from all of the rooms to be served by each return-air intake.

10. From Fig. 69 and reference to the basement layout determine which of the illustrated typical arrangements most nearly coincides with each of the return-air conduits from register to furnace shoe.

11. The sum of the heat losses and the letter representing the arrangement type are used with Table 40 to find the return-air combination number to be used in connection with each return-air intake.

12. The return-air combination number obtained from Table 40 is used in connection with Table 41 to determine the actual dimensions of the various parts of each return-air arrangement selected.

13. Select a furnace having a register delivery rating, in Btu per hour, equal to the sum of the calculated heat losses from all the rooms to be served.

TABLE 40

GRAVITY-FLOW WARM-AIR SYSTEMS, RETURN AIR CARRYING CAPACITY, BTU SERVICED PER HOUR

Return-Air Combination Number	Duct Diameter, In.	Type *A*, Btu per Hr	Types *B* and *C*, Btu per Hr	Type *D*, Btu per Hr	Type *E*, Btu per Hr	Type *F*, Btu per Hr	Return-Air Combination Number
31	10	11,300	9,500	7,800	5,000	7,800	31
32	12	16,300	13,700	11,300	7,200	11,300	32
33	14	22,200	18,700	15,300	9,800	15,300	33
34	16	29,000	24,400	20,000	12,800	20,000	34
35	18	36,700	30,800	25,300	16,200	25,300	35
36	20	45,300	38,000	31,300	20,000	31,300	36
37	22	54,800	46,000	37,800	24,100	37,800	37
38	24	65,200	54,800	45,000	28,700	45,000	38

Although the return air as such, does not contribute toward room heating, the volume to be handled by any arrangement of return-air register, duct, and shoe is proportional to the heat losses from all the rooms served by it. The total heat loss from the structure is always less than the sum of the heat losses from all the individual rooms because infiltration of cold air does not occur in all the rooms at the same time. Therefore, in selecting a furnace according to item 13 of the outlined procedure, some reserve capacity is provided.

Figure 70 shows the recommended practice in joining two horizontal return ducts to a common vertical or inclined duct. Four different types of shoes for joining a vertical or an inclined duct to a furnace casing are illustrated by Fig. 71.

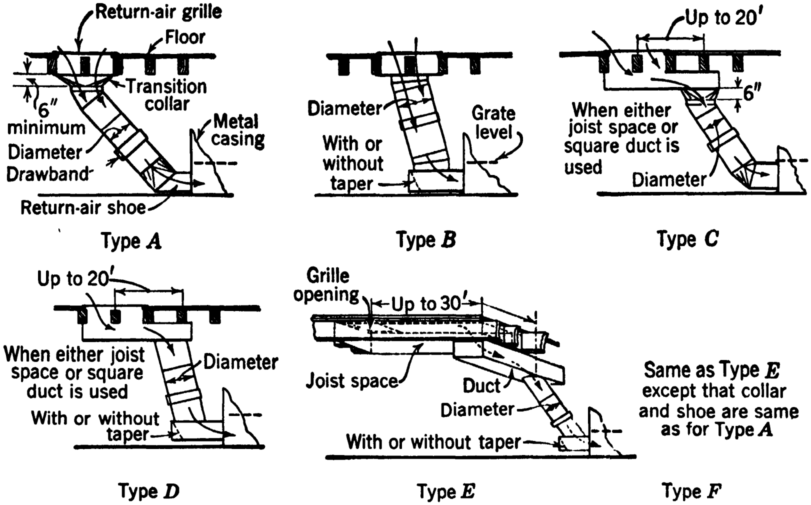
TABLE 41

Gravity-Flow Warm-Air Systems, Return-Air Ducts, Combinations of Parts Selected as Standard

Return Air Combination Number	Duct Diameter	Duct Area, Sq In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes, In. Choose One			Baseboard Intake for First Story	When Joist Lining is Used,* Number of Joist Spaces Lined and Minimum Depth of Space Required†	When Duct is Used. Choose One	
				a	*b*	*c*				
31	10	78			8 × 14	10 × 12		1 space of 7.0-in. depth	14 × 6	12 × 8
32	12	113		6 × 30	8 × 24	12 × 14	14 × 10 4-in. Extension	1 space of 9.0 in. depth or 2 spaces of 5.0-in. depth	28 × 6	16 × 8
33	14	154	170	8 × 30	10 × 24	14 × 16		1 space of 12.0-in. depth or 2 spaces of 6.0-in. depth	28 × 6	22 × 8
34	16	201	220	10 × 30	12 × 24		30 × 10 4-in. Extension	1 space of 16.0-in. depth or 2 spaces of 8.0-in. depth	28 × 8	22 × 10
35	18	254	280	12 × 30	14 × 24			2 spaces of 10.0-in. depth	36 × 8	28 × 10
36	20	314	340	14 × 30	18 × 24			2 spaces of 12.5-in. depth	36 × 10	30 × 12
37	22	380	420	18 × 30				2 spaces of 15.0-in. depth	42 × 10	36 × 12
38	24	452	500	20 × 30				2 spaces of 18.0-in. depth	42 × 12	36 × 14

* Based on 14-in. space between joists.

† Use full depth of joist except when joist depth is less than minimum depth required, when pan must be used.



Note: For types *C, D, E,* and *F,* return-air duct systems reduce the carrying capacities shown in Table 40 by one per cent for each 4 ft additional length in the horizontal run.

FIG. 69. Arrangements of gravity-flow return-air duct systems.

Table 42 is a work-sheet tabulation used in applying the outlined procedure to the two-story house for which the heating plan of Fig. 66 was made.

One return-air register serves the entire house in the heating plan of Fig. 66. The sum of the heat losses of the rooms served is 33,745 Btu per hr, the arrangement is type *A*, referring to Fig. 69, and the

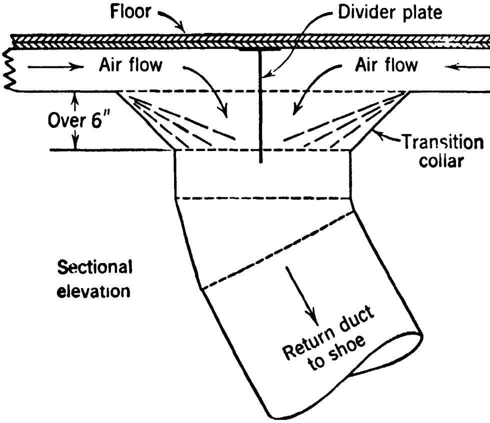


FIG. 70. Recommended connection of two horizontal gravity-flow return-air ducts.

proper combination number was found from Table 40 to be number 35. From Table 41 it was found that combination 35 consists of an 18-in. diameter recirculating duct, either a 14 × 24 in. or a 12 × 30 in. metal grille, and a shoe having an area of 280 sq in. at the connection to the furnace casing. The shape of the 14 × 24 in. register is better adapted to the location chosen for it on the plan of Fig. 66.

In considering the arrangement of the rooms, the layout of the basement, and the direction of the joists in the plan of Fig. 66 it seemed advisable to use only one return-air register. It is unlikely that noticeable drafts would occur in this house because the walls are insulated, and storm windows are provided at all window openings. Where there are one or more large areas of single glass such as picture windows, additional return-air registers should be located as close to them as practicable so as to carry cold air from them to the furnace so that it will not cross the floor of the room.

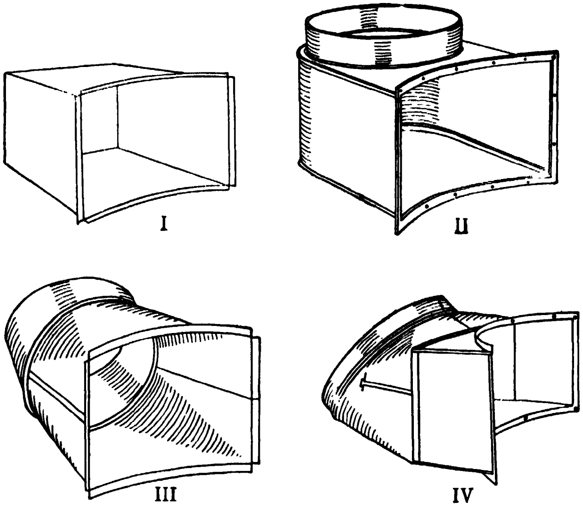


FIG. 71. Recirculating-duct shoes.

141. Design of Forced-Circulation Warm-Air Furnace Systems. The air-carrying ducts of large systems or systems serving buildings of three or more stories should be designed by the method which is outlined in Art. 302. For all heating plants of this type, serving one- or two-story buildings which have design heat losses less than 150,000 Btu per hour, the following procedure, similar to the one previously given for gravity-flow plants in the preceding article, is recommended by the same authority. Tables 44, 45, 46, and 48, which may be

TABLE 42

Work Sheet Illustrating the Method Used in Sizing the Warm-Air Leaders for the Plans of Fig. 66

1. Room use	Living*	Dining	Kitchen	Hall	Bedroom 1	Bedroom 3	Bath	Bedroom 2	Den
2. Story	1	1	1	1	2	2	2	2	2
3. Total heat loss, Btu per hr	N 3270 S 3270	3175	3150	5170	4365	3380	1560	3330	3075
4. Length of leader, bonnet to boot, ft	N 6 S 4	14	10	4	4	9	7	2	9
5. Number of elbows	N 1 S 2	2	4	1	2	2	1	1	3
6. Combination number	N 1 S 1	1	1	1	11	11	11	11	11
7. Leader diameter, in.	N 8 S 8	8	8	8	8	8	8	8	8
8. Stack or throat size, in.	$6\frac{1}{8} \times 10$	$6\frac{1}{8} \times 10$	$6\frac{1}{8} \times 10$	$6\frac{1}{8} \times 10$	$10 \times 3\frac{1}{4}$	$10 \times 3\frac{1}{4}$	$10 \times 3\frac{1}{4}$	$10 \times 3\frac{1}{4}$	$10 \times 3\frac{1}{4}$
9. Register type and size, in.	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8	Baseboard 10 × 8

* Two registers are provided in the living room.

used for determining the required dimensions for the trunks, branch pipes, stacks and registers, are based on an assumed temperature rise through the furnace of 100 F under design conditions and an assumed friction pressure loss of 0.10 in. of water between the furnace bonnet and the register outlets and between the return-air inlets and the blower cabinet. The various steps in the design procedure are as follows:

1. Locate all supply and return-air intakes on the first- and second-floor plans of the house to be heated (see Fig. 72 and Art. 139).

2. Indicate any required crossover connection between a second-floor register and the stack location in a first-floor partition where the building construction does not permit the normal arrangement. No crossover is required in the plan of Fig. 72.

3. On a basement plan similar to the one of Fig. 72 show the location of the following items:

a. Warm-air and return-air stacks for first- and second-floor rooms. The numbers and letters at the stack locations on the basement plan of Fig. 72 indicate the type of fitting used with reference to Fig. 73.

b. Location of furnace showing smokepipe from it to the chimney. If the fan is in a separate cabinet show its location beside the furnace. The fan in the system of Fig. 72 is located in the lower part of the furnace casing as shown in Fig. 58.

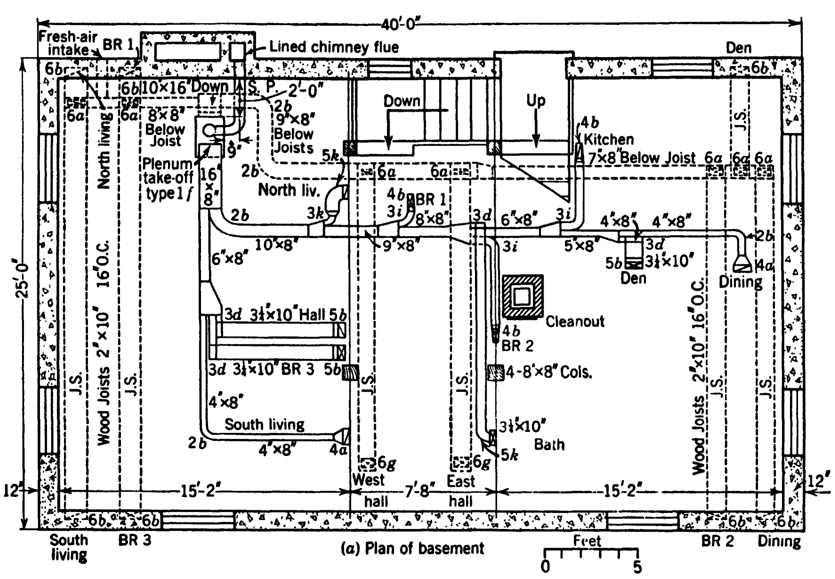
c. The system of ducts which connects the plenum at the top of the furnace to the stacks leading to the various registers. Use solid lines to indicate this system of ducts (see Fig. 72).

d. Return-air system of ducts connecting return-air stacks or boots to the return-air plenum. Use broken lines to indicate this system of ducts (see Fig. 72).

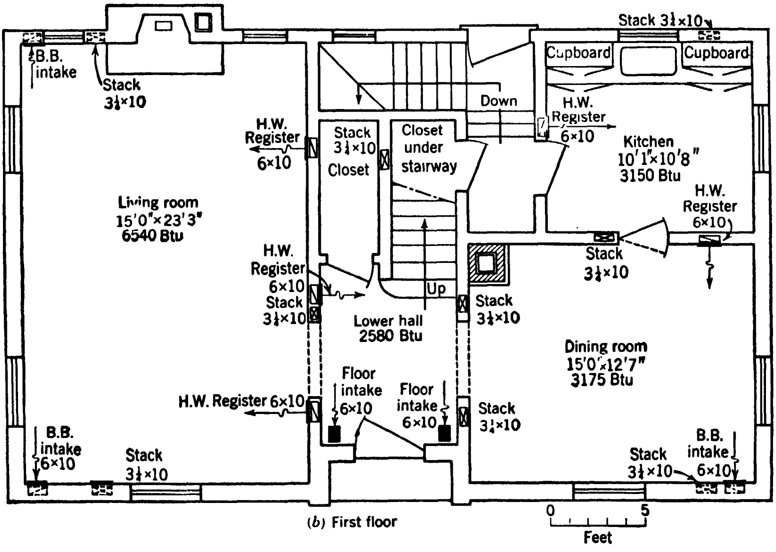
4. Calculate the heat losses from each room, and insert as item 3 in a table similar to Table 43.

5. From the basement plan determine the actual length from the bonnet to the boot for each branch that is to deliver warm air and enter this figure as item 4 in the table. Use two columns for one room if it is served by two different branches. Include the horizontal length of any crossovers in a joist space between first and second floor, but do not include vertical lengths of stacks.

6. Determine the equivalent length of fittings and register for the system serving each register (see Fig. 73). The equivalent length of a

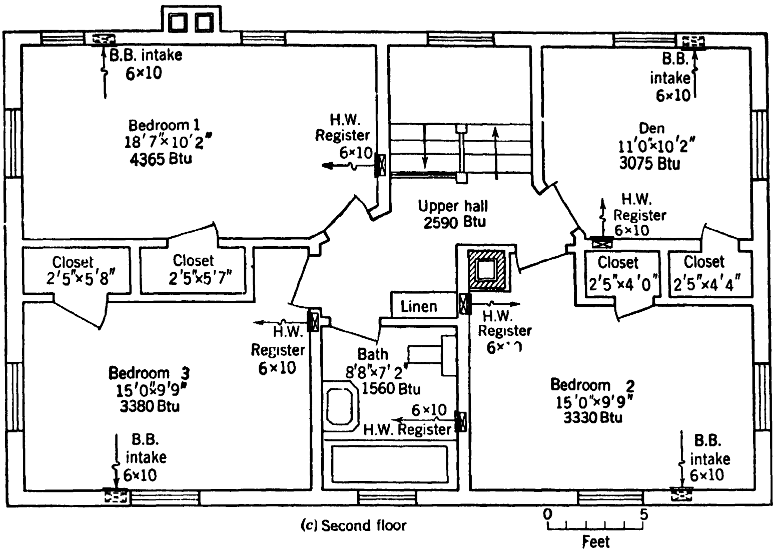


(a) Plan of basement



(b) First floor

Fig. 72. *(Continued on page 185.)*



(c) Second floor

FIG. 72. Layout of a forced-circulation warm-air heating system in a residence.

bonnet take-off is included in the total, for every branch duct served when a trunk serves several branches. Similarly the equivalent length of an elbow in a trunk is included in the total for all branch ducts served by trunk take-offs located downstream from such a fitting. Enter the sum of these equivalent lengths as item 6 in a table such as Table 43. The equivalent lengths of the fittings in that portion of the system which serves the kitchen in the plan of Fig. 72 are as follows: plenum take-off, type *f*, 35 ft; elbow in trunk, 10 ft; trunk take-off, type *i*, 10 ft; boot, type *b*, 35 ft; register with 22-deg deflection angle, 45 ft; total, 135 ft. Item 6 in Table 43 was calculated for each of the other rooms in the same manner.

7. Using Table 44 for a first-story room, or Table 45 for a second-story room, and the information given in items 3, 4, and 6 of Table 43, read the proper combination number at the right side of the table used, and enter this number as item 7 in Table 43. The combination selected is the one for which the table shows a Btu capacity closest to the estimated heat losses from the room to be heated.

8. Using the combination number which has been selected determine the dimensions of the stack, branch pipe, register, and the required increase in trunk width from Table 46, and enter these data as items 8, 9, 10, and 11 in a tabulation similar to Table 43.

9. After proper dimensions of all warm-air branch ducts have been

TABLE 43

WORK SHEET ILLUSTRATING THE METHOD USED IN SIZING THE WARM-AIR DUCTS FOR THE PLAN OF FIG. 72

1. Room use	N $\frac{1}{2}$ of Living*	S $\frac{1}{2}$ of Living	Dining	Kitchen	Upper and Lower Hall	Bed-room 1	Bed-room 3	Bath	Bed-room 2	Den
2. Story	1	1	1	1	1	2	2	2	2	2
3. Heat loss, Btu per hr	3270	3270	3175	3150	5170	4365	3380	1560	3330	3075
4. Horizontal length, furnace to register, ft	12	22	33	27	16	13	17	29	23	28
5. Type and deflection angle of register, deg.	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22	High wall 22
6. Equivalent length† of fittings and register, ft	160	120	130	135	125	135	125	180	135	135
7. Combination number	41	41	41	41	41	41	41	41	41	41
8. Stack size, in.	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$	$3\frac{1}{4} \times 10$
9. Branch-duct size, in.	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8	4 × 8
10. Register dimensions, in.	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10	6 × 10
11. Required increase in trunk width, in.	1	1	1	1	1	1	1	1	1	1

* Two registers used in this room.

† Includes equivalent length of bonnet take-off, elbows in trunk if any, trunk take-off, boot from branch to stack, elbows in branch or stack if any, and register.

TABLE 44

CAPACITY TABLES FOR FORCED-CIRCULATION, WARM-AIR AND RETURN-AIR BRANCHES, FIRST STORY

For Uninsulated Metal Ducts	Actual Length from Bonnet to Boot or from Return Plenum to Boot, Ft								Warm-Air Combination Number	Return-Air Combination Number
	1 to 7 ft	8 to 12 ft	13 to 17 ft	18 to 24 ft	25 to 34 ft	35 to 44 ft	45 to 54 ft	55 to 64 ft		
	Column *a*	Column *b*	Column *c*	Column *d*	Column *e*	Column *f*	Column *g*	Column *h*		
Section *A* 40 to 69 equivalent feet for fittings and register	7,200	6,700	6,100	5,600	4,800	4,100	3 500	3,000	41	51
	12,500	11,700	10,800	9,900	8,500	7,400	6,400	5,500	42	52
	16,000	15,000	14,000	13,000	11,300	9,900	8,700	7,700	43	53
	19,100	18,000	17,000	16,000	14,200	12,500	11,000	9,600	44	54
	25,000	23,400	21,600	19,800	17,000	14,800	12,800	11,000	45*	55
	32,000	30,000	28,000	26,000	22,600	19,800	17,400	15,400	46*	56
	80,000	75,000	70,000	65,000	56,500	49,500	43,500	38,500	...	57*
Section *B* 70 to 99 equivalent feet	5,500	5,100	4,800	4,500	3,900	3,400	3.000	2,600	41	51
	9,900	9,200	8,600	8,100	7,100	6,200	5,400	4,600	42	52
	13,100	12,300	11,600	10,900	9,700	8,500	7,500	6,500	43	53
	16,300	15,400	14,500	13,700	12,200	10,800	9,500	8,300	44	54
	19,800	18,400	17,200	16,200	14,200	12,400	10,800	9,200	45*	55
	26,200	24,600	23,200	21,800	19,400	17,000	15,000	13,000	46*	56
	65,500	61,500	58,000	54,500	48,500	42,500	37,500	42,500	...	57*
Section *C* 100 to 139 equivalent feet	4.400	4,200	3,900	3,700	3,200	2,900	2,600	2,300	41	51
	8,100	7,600	7,100	6,600	5,800	5,100	4,500	4,000	42	52
	10,800	10,100	9,600	9,000	8,000	7,100	6,200	5,400	43	53
	13,700	12,900	12,100	11,300	10,000	8,900	7,900	7,000	44	54
	16,200	15,200	14,200	13,200	11,600	10,200	9,000	8,000	45*	55
	21,600	20,200	19,200	18,000	16,000	14,200	12,400	10,800	46*	56
	54,000	50,500	48,000	45,000	40,000	35,500	31,000	27,000	...	57*
Section *D* 140 to 189 equivalent feet	3,900	3,700	3,500	3,300	2,900	2,500	2,200	2,000	41	51
	6,900	6,500	6,100	5,700	5,100	4,500	4,000	3,600	42	52
	9,300	8,700	8,100	7,600	6,800	6,000	5,300	4,700	43	53
	11,600	11,000	10,400	9,800	8,700	7,700	6,800	6,000	44	54
	13,800	13,000	12,200	11,400	10,200	9,000	8,000	7,200	45*	55
	18.600	17,400	16,200	15,200	13,600	12,000	10,600	9,400	46*	56
	46,500	43,500	40,500	38,000	34,000	30,000	26,500	23,500	...	57*
Section *E* 190 to 250 equivalent feet	3,300	3,100	2,900	2,700	2,500	2,200	1,900	1,700	41	51
	5,700	5,400	5,100	4,800	4,300	3,800	3,400	3,000	42	52
	7,800	7,300	6,900	6,500	5,700	5,000	4,400	3,900	43	53
	9,700	9,100	8,600	8,100	7,300	6,500	5,800	5,100	44	54
	11,400	10,800	10,200	9,600	8,600	7,600	6,800	6,000	45*	55
	15,600	14,600	13,800	13,000	11,400	10,000	8,800	7,800	46*	56
	39,000	36,500	34,500	32,500	28,500	25,000	22,000	19,500	...	57*
For insulated ducts	Column *a* 1 to 9 ft	Column *b* 10 to 17 ft	Column *c* 18 to 24 ft	Column *d* 25 to 34 ft	Column *e* 35 to 54 ft	Column *f* 55 to 74 ft	For ducts that are *completely insulated* with ½ in. thick insulation from *bonnet to boot*, use these column headings			

* Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

These tables are for use in sizing *both the warm-air and the return-air* branches.

Frictional resistances and temperature drops in ducts have both been accounted for in these tables.

TABLE 45

Capacity Tables For Forced-Circulation, Warm-Air and Return-Air Branches, Second Story

For Uninsulated Metal Ducts	Actual Length from Bonnet to Boot or from Return Plenum to Boot, Ft								Warm-Air Combination Number	Return-Air Combination Number
	1 to 7 ft	8 to 12 ft	13 to 17 ft	18 to 24 ft	25 to 34 ft	35 to 44 ft	45 to 54 ft	55 to 64 ft		
	Column *a*	Column *b*	Column *c*	Column *d*	Column *e*	Column *f*	Column *g*	Column *h*		
Section *A* 40 to 69 equivalent feet for fittings and register	6,300	5,700	5,200	4,800	4,100	3,500	3,100	2,800	41	51
	10,900	10,000	9,200	8,500	7,300	6,400	5,600	5 000	42	52
	14,000	13,000	12,100	11,400	10 000	8,800	7,800	6,800	43	53
	17,000	15,900	14,900	13,900	12,400	11,100	9,900	9,000	44	54
	21,800	20,000	18,400	17,000	14,600	12,800	11,200	10,000	45*	55
	28,000	26,000	24,200	22,400	20,000	17,600	15,600	13,600	46*	56
	70,000	65,000	60,500	57,000	50,000	44,000	39,000	34 000	...	57*
Section *B* 70 to 99 equivalent feet	5,000	4,600	4,300	4,000	3,400	3,000	2,700	2,300	41	51
	9,000	8,200	7,600	7,100	6,200	5,400	4,700	4,200	42	52
	11,900	11,000	10,300	9,600	8,400	7,500	6,700	6,000	43	53
	14,800	13,900	13,000	12,200	10,800	9,600	8,500	7,500	44	54
	18,000	16,400	15,200	14,200	12,400	10,800	9,400	8,400	45*	55
	23,900	22,000	20,600	19,200	16,800	15,000	13,400	12,000	46*	56
	59,500	55,000	51,500	48,000	42,000	37,500	33,500	30,000	...	57*
Section *C* 100 to 139 equivalent feet	4,100	3,800	3,600	3,400	2,900	2,600	2,300	2,000	41	51
	7,400	6,900	6,400	6,000	5,200	4,600	4,000	3,600	42	52
	10,000	9,300	8,700	8,100	7,100	6,200	5,500	4,800	43	53
	12,500	11,600	10,800	10,100	9,000	8,000	7,200	6,500	44	54
	14,800	13,800	12,800	12,000	10,400	9,200	8,000	7,200	45*	55
	20,000	18,600	17,400	16,200	14,200	12,400	11,000	9,600	46*	56
	50,000	46,500	43,500	40,500	35,500	31,000	27,500	24,000	...	57*
Section *D* 140 to 189 equivalent feet	3,700	3,400	3,100	2,900	2,600	2,200	2,000	1,800	41	51
	6,500	6,000	5,600	5,300	4,700	4,100	3,700	3,300	42	52
	8,600	8,000	7,500	7,000	6,100	5,400	4,800	4,300	43	53
	10,900	10,100	9,400	8,800	7,800	6,900	6,100	5,400	44	54
	13,000	12,000	11,200	10,600	9,400	8,200	7,400	6,600	45*	55
	17,200	16,000	15,000	14,000	12,200	10,800	9,600	8,600	46*	56
	43,000	40,000	37,500	35,000	30,500	27,000	24,000	21,500	...	57*
Section *E* 190 to 250 equivalent feet	3,200	2,900	2,700	2,500	2,200	1,900	1,700	1,500	41	51
	5,500	5,200	4,800	4,400	3,900	3,400	3,100	2,800	42	52
	7,300	6,800	6,300	5,900	5,100	4,500	3,900	3,500	43	53
	9,100	8,500	7,900	7,500	6,600	5,900	5,200	4,700	44	54
	11,000	10,400	9,600	8,800	7,800	6,800	6,200	5,600	45*	55
	14,600	13,600	12,600	11,800	10,200	9,000	7,800	7,000	46*	56
	36,500	34,000	31,500	29,500	25,500	22,500	19,500	17,500	...	57*
For insulated ducts	Column *a* 1 to 8 ft	Column *b* 9 to 14 ft	Column *c* 15 to 20 ft	Column *d* 21 to 27 ft	Column *e* 28 to 42 ft	Column *f* 43 to 54 ft	Column *g* 55 to 70 ft	For ducts that are *completely insulated* with ½ in. thick insulation from *bonnet to boot*, use these column heading		

* Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

These tables are to use in sizing *both the warm-air and the return-air* branches.

Frictional resistance and temperature drops in ducts have both been accounted for in these tables.

TABLE 46

FORCED-CIRCULATION WARM-AIR DUCT SYSTEMS, COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination Number	Stack Size, In.	Branch-Pipe Size, In.		Register Size, In.				Required Increase of Width of Trunk Duct, In.
		Round	Rectangular	Baseboard or Low Sidewall	High Sidewall		Floor Registers*	
					For Throws Less Than 13 Ft	For Throws More Than 13 Ft		
1	2	3	4	5	6	7	8	9
41	10 × 3¼	6	4 × 8	10 × 6	10 × 6	10 × 4	8 × 10*	1
42	10 × 3¼	6	4 × 8	10 × 6	10 × 6	10 × 4	8 × 10*	2
43	12 × 3¼	7	5 × 8	12 × 6	12 × 6	12 × 4	9 × 12*	3
44	14 × 3¼	8	6 × 8	14 × 6	14 × 6	14 × 4	9 × 12* or longer	4
45	10 × 3¼ (2 stacks)	9	8 × 8	(2) 10 × 6 or (1) 24 × 6	(2) 10 × 6 or (1) 24 × 6	(2) 10 × 4 or (1) 24 × 4	10 × 12*	6
46	12 × 3¼ (2 stacks)	10	10 × 8	(2) 12 × 6 or (1) 30 × 6	(2) 12 × 6 or (1) 30 × 6	(2) 12 × 4 or (1) 30 × 4	12 × 14*	7

* Use these items only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm-air furnace systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested.

determined the warm-air trunk ducts may be sized in all their parts as follows:

a. Assume that the most remote branch duct is the end of the trunk to which it is attached.

b. The trunk width increases between the discharge end and the furnace air outlet. After the duct juncture with each branch the amount of width increase is as shown by item 11 of Table 43. For example, the most remote section of the longest trunk duct used in the plan of Fig. 72 is 4 × 8 in., which is the branch-duct size required to serve the dining room. The branch duct serving the den calls for a 1-in. extension in the

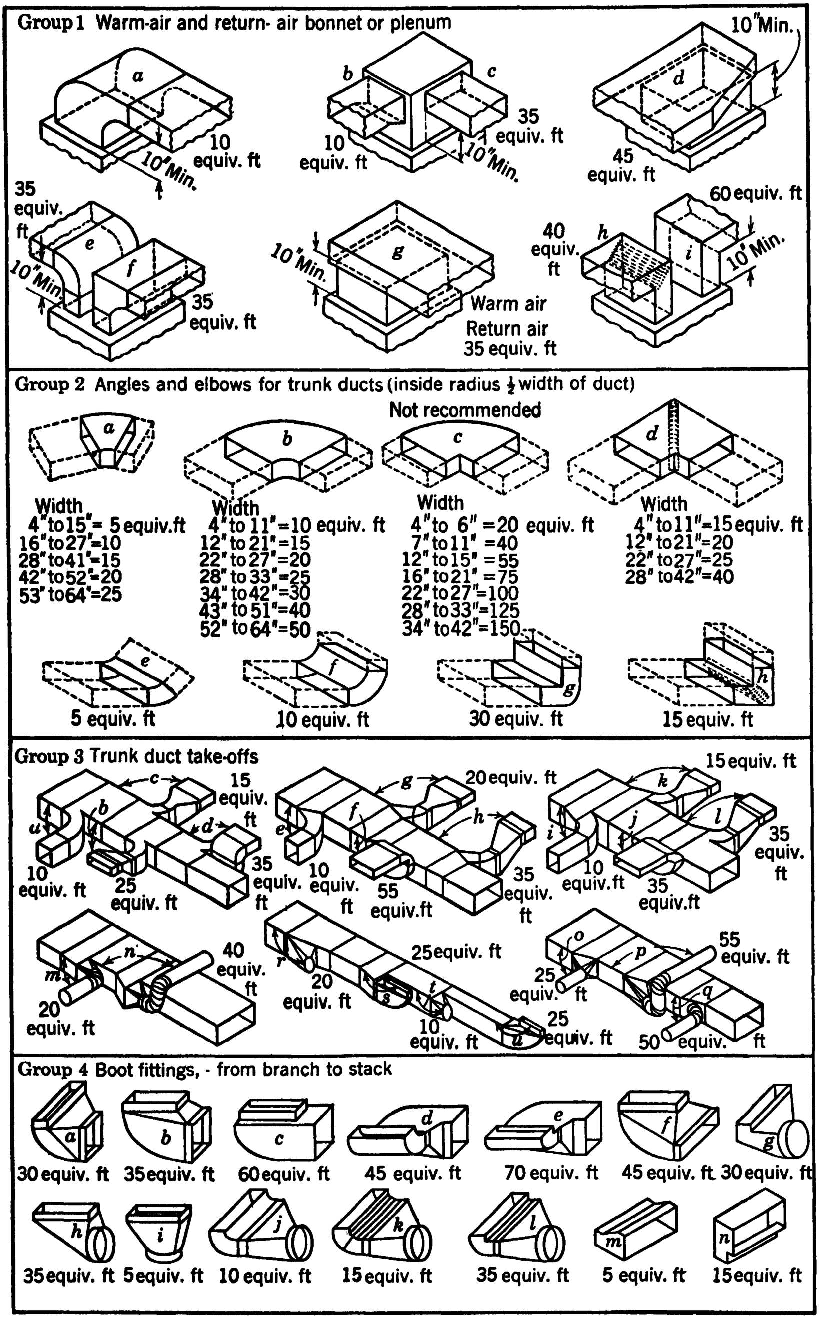
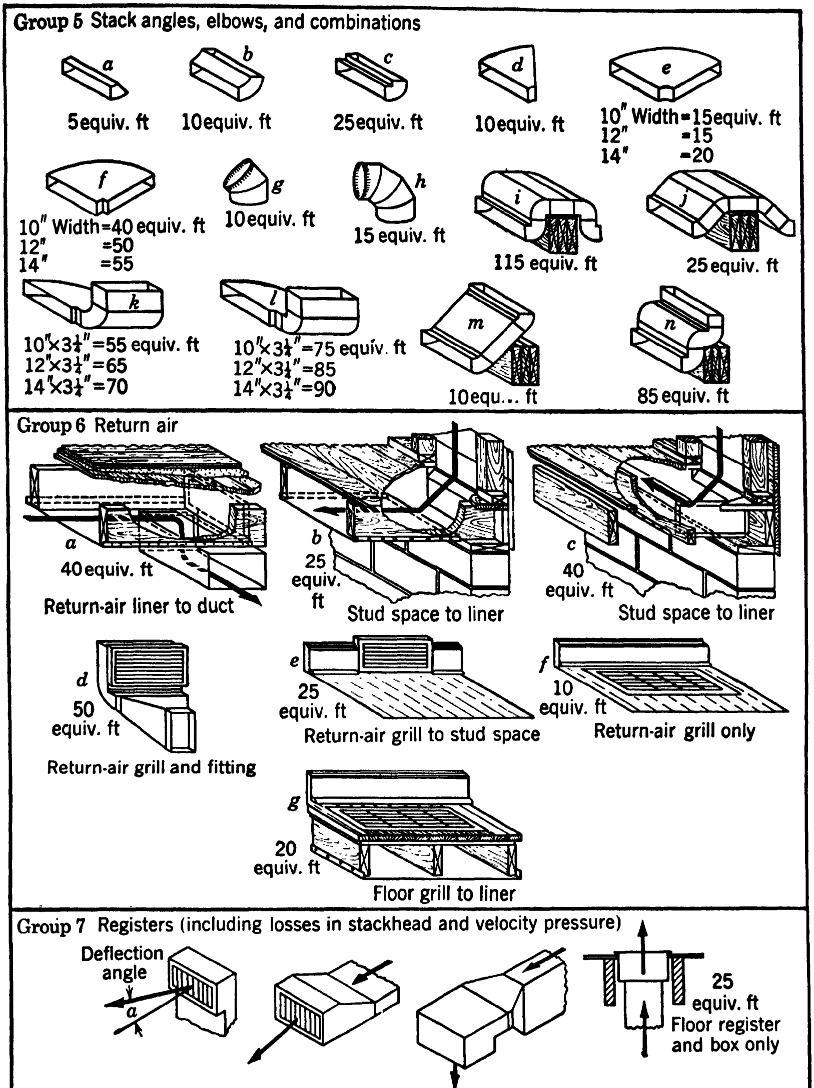


Fig. 73*a*. Equivalent lengths of fittings.



Equivalent length for registers

Deflection angle *a*	0°	15°	22°	30°	45°
Baseboard, high or low sidewall register, equiv. ft	35	40	45	60	115
Floor register with box only, equiv. ft	25				

For two-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle *a* in table.

Fig. 73*b*. Equivalent lengths of fittings.

trunk width, so the trunk dimensions immediately upstream from the juncture with this branch are 5 × 8 in. In a similar manner the width of the trunk is increased by the proper amount upstream from each point where a branch is attached. In the layout of Fig. 72, the duct between the junction of the two trunk ducts and the furnace is considered as an extension of the 16-in.-wide plenum chamber.

10. Note the location of return-air registers on first- and second-floor plans, and fill in items 1 and 2 in a table having the form of Table 47.

11. Enter as item 3 the total heat loss from the space or spaces to be served by each return-air intake.

12. From the basement plan of the system, determine the actual length from each return-air register or stack to the return-air plenum, and enter these data as item 4 in the same table.

13. From Fig. 73 determine the equivalent length of the fittings only in each part of the return-air system, and enter as item 5 in the table. The equivalent length of each of the fittings in the path of the return air which enters the system through the register located in the dining room is as follows: return intake to stud space, 6e Fig. 73, 25 ft; connection between stud space and lined joist space, 6*b* Fig. 73, 25 ft; connection between lined joist space and trunk duct, 6*a* Fig. 73, 40 ft; two elbows in return air trunk, 2*b* Fig. 73, each 10 ft; connection between return-air trunk and return-air plenum, assumed to be equivalent to 1*f* Fig. 73, 35 ft; and the elbow at the bottom of the plenum which directs the air into the lower part of the furnace, assumed to be equivalent to 2*f* Fig. 73, 10 ft; total equivalent length of all fittings 155 ft. In a similar manner the equivalent length of the fittings in each branch of the return-air system of Fig. 72 was determined and the amounts listed as item 5 in Table 47.

14. With the data given in items 2, 3, 4, and 5 of a table similar to 47 and using either Table 44 or Table 45 determine the number of the proper return-air combination for serving each return-air intake, and enter as item 6 in the table.

15. Using the combination number from step 14, fill in items 7, 8, 9, 10, and 11 in the form of Table 47 by referring to the proper line in Table 48.

16. After a table such as Table 47 has been completed, the return-air trunk ducts may be sized by the employment of the same procedure that was used in step 9 for sizing the warm-air trunks. However, it is not necessary to change the size of a return-air trunk duct between adjacent branches feeding into it. It is often possible to simplify

TABLE 47

WORK SHEET ILLUSTRATING THE METHOD USED IN SIZING THE RETURN-AIR DUCTS FOR THE PLAN OF FIG 72

1. Location of intake	North $\frac{1}{2}$ of Living	South $\frac{1}{2}$ of Living	Bedroom 1	Bedroom 3	West Hall*	East Hall*	Bedroom 2	Den	Dining	Fresh-Air Intake†
2. Story	1	1	2	2	1	1	2	2	1	Basement
3. Heat loss from space or spaces served, Btu per hr	3270	3270	4365	3380	3365	3365	3330	3075	3175	3150 †
4. Horizontal distance intake to furnace, ft	9	29	6	26	27	32	49	36	52	8
5. Equivalent length of fittings, ft	135	135	135	135	125	125	155	155	155	95
6. Combination number	51	51	51	51	51	51	52	51	52	51
7. Stack size, in.			$3\frac{1}{4}$ × 10	$3\frac{1}{4}$ × 10			$3\frac{1}{4}$ × 10	$3\frac{1}{4}$ × 10		
8. Branch-duct size, in.										6 in. round
9. Number of joists and depth required‡	1–3 in.	1–3 in.	1–3 in.	1–3 in.	1–3 in.	1–3 in.	1–3 in.	1–3 in.	1–3 in.	
10. Intake type and size, in.	Baseboard 6 × 10	Baseboard 6 × 10	Baseboard 6 × 10	Baseboard 6 × 10	Floor 6 × 10	Floor 6 × 10	Baseboard 6 × 10	Baseboard 6 × 10	Baseboard 6 × 10	Special 6 × 10 equivalent
11. Required increase in trunk width, in.	1	1	1	1	1	1	2	1	2	1

* Two return registers located by the front entrance in the lower hall to serve both upper and lower hall together with the bathroom.

† Fresh-air intake is proportioned to replace the air which is delivered to the kitchen and not returned.

‡ Depths shown in table are minimum, use full depth of joist spaces.

TABLE 48

FORCED-CIRCULATION WARM-AIR RETURN-DUCT SYSTEMS, COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination Number	Return-Air Intake Size, In.		Riser Size in Inches Where Stack Is Used in Stud Space	Branch Pipe Size, In.		When Joist Lining Is Used,† Number of Joist Spaces Lined and Minimum Depth of Space Required	Required Increase in Width of Trunk Duct, for 8-In. Depth of Duct, In.
	Baseboard	Floor*		Round	Rectangular		
1	2	3	4	5	6	7	8
51	10 × 6	6 × 10 or 4 × 14	10 × 3¼‡	6	4 × 8	1 space of 3-in. depth	1
52	10 × 6	6 × 10 or 4 × 14	10 × 3¼‡	6	4 × 8	1 space of 3-in. depth	2
53	12 × 6	6 × 12 or 6 × 14	12 × 3¼§	7	5 × 8	1 space of 4-in. depth	3
54	14 × 6	6 × 14	14 × 3¼§	8	6 × 8	1 space of 5-in. depth	4
55	24 × 6 or 30 × 6	6× 30	Two stacks 10 × 3¼‡	9	8 × 8	1 space of 6-in. depth or 2 spaces of 3-in. depth	6
56	30 × 6	6 × 30	Two stacks 12 × 3¼§	10	10 × 8	1 space of 7-in. depth or 2 spaces of 4-in. depth	7
57	. . .	8 × 30	. . .	12	15 × 8	1 space of 9-in. depth or 2 spaces of 5-in. depth	11

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* Use these items only when building construction, or capacities, requires the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

† Based on 14-in. space between joists. Use full depth of joist, except when joist depth is less than minimum depth required, in which case a drop pan must be used. This may occur when two or more return ducts are connected to the same joist space.

‡ If it is desired to use 14-in. × 3⅝-in. stud space, it makes no difference whether this space has protruding keys or not.

§ If it is desired to use 14-in. × 3⅝-in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

the construction by continuing a trunk duct at a uniform size past the junction with two or more branches. Where the dimensions are not changed, the duct size used must be that which is required at the end handling the greater volume of air. In the system of Fig. 72 the most remote section of the longest return-air trunk was sized as follows: The most remote branch which serves the dining room is a combination 52, which requires a branch size 4 × 8 in., and this is the required size of the trunk at the point where it receives the air from the joist space used to conduct it from the register and stud space. The branch from the den is combination 51, which requires a 1-in. extension of the trunk width, and the branch from bedroom 2 is combination 52, which calls for a 2-in. extension. The required trunk width downstream from the juncture with the branch serving bedroom 2 is therefore $4 + 1 + 2 = 7$ in. To facilitate fabrication, the trunk dimensions are specified as 7 × 8 in. from the most remote end to a point upstream from the junction with the branch from the east hall. At this point the width is increased 2 in. in one step to accommodate the flow from the other two branches.

17. Select a furnace having a register delivery rating in Btu per hour which is equal to the sum of the heat losses from all of the rooms to be heated. As for a gravity-flow system, selection of a furnace on this basis will result in the installation of a furnace having some reserve capacity.

TABLE 49*

SUGGESTED MINIMUM BLOWER SIZE FOR SEPARATE BLOWER INSTALLATIONS

Total Heat Losses, Btu per Hr	Minimum Blower Wheel Diameter,† In.	Minimum Motor Size,‡ Hp	Required Air Delivery, Cfm	Select Speed to Secure Static§ Pressure of, In. of Water
1	2	3	4	5
Up to 40,000	9	$\frac{1}{6}$	520	$\frac{3}{8}$
40,000 to 60,000	9	$\frac{1}{6}$	700	$\frac{3}{8}$
60,000 to 80,000	10	$\frac{1}{6}$	950	$\frac{3}{8}$
80,000 to 120,000	12	$\frac{1}{4}$	1400	$\frac{3}{8}$
120,000 to 150,000	14	$\frac{1}{3}$	1770	$\frac{3}{8}$

* This table applies only to installations in which the blower is selected separately from the furnace. It *does not apply* to packaged units in which the blower is integrally assembled with the furnace.

† Based on multi-blade, forward-curved blade, double-width, double-inlet type, or equivalent.

‡ A long-hour duty type of motor is required.

§ Blower speeds should be adjustable approximately 10 per cent faster and 10 per cent slower than the speed required to maintain the listed static pressures.

18. If the blower is not furnished by the furnace manufacturer, Table 49 may be used as a guide in the selection of a fan.

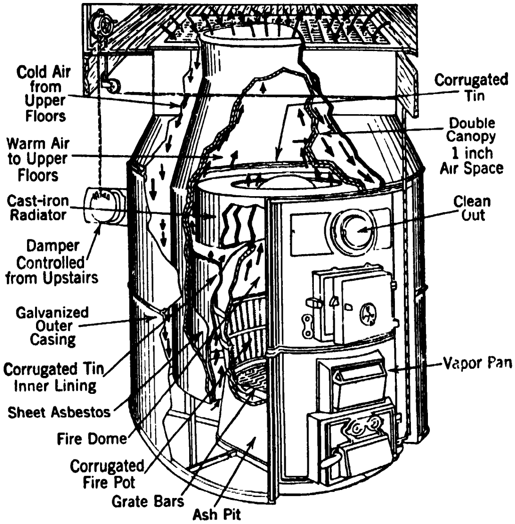


Fig. 74. Pipeless warm-air furnace.

142. Pipeless Warm-Air Furnaces. Installations of single-pipe or pipeless furnaces have a limited field in small residences with a single floor and in buildings with open spaces as stores and small churches.

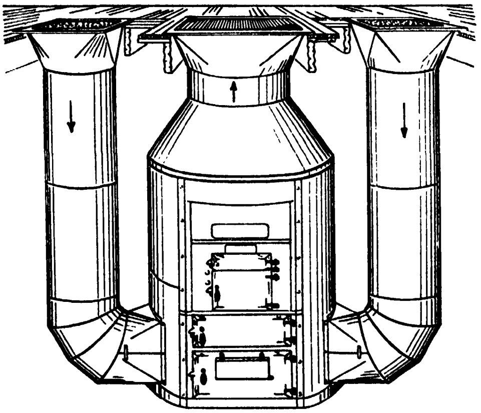


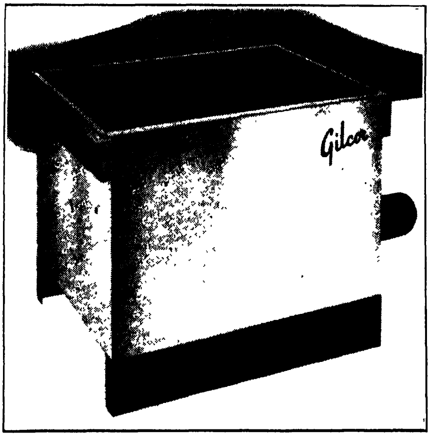
Fig. 75. Three-way gravity warm-air furnace.

The furnace illustrated, Fig. 74, has separate inner and outer casings and a register common to both the supply and return air connections. The return air enters the outer portion of the grille and flows downward between the casings to the bottom of the inner lining. The air then passes upward through the inner casing to move over the heating surfaces of the furnace. The warmed air leaves the furnace through a very short vertical duct leading to the floor register where it enters the room through the central portion of the floor register. The supply and return connections beneath the grille are entirely separated. Installations are also made with separate outlet and return connections as in Fig. 75.

Most commercial furnaces can be installed as pipeless units if the casing is properly designed. The important item in casing design is the return of cool air to the bottom of the casing without its being appreciably warmed. Warming the return air before it enters the inner casing reduces the motive head of the plant. Those furnaces having a single warm-air outlet, and the form of cold-air returns of Fig. 75 function somewhat better than those arranged in the manner illustrated by Fig. 74.

The first costs of installation are lower than for a piped furnace, and operating costs are about the same. Difficulty may be encountered in the maintenance of uniform temperature conditions owing to limited air circulation in the heated space, and also there is a tendency toward excessive stratification of the air in the space into which the warmed air is introduced.

143. Floor Furnaces. A special type of pipeless warm-air furnace, known as a floor furnace, is used extensively for heating one-story houses, built without basements, in certain sections of the country. Oil and gas are the common fuels used with floor furnaces which are set in floor openings, as near as possible to the center of the floor area of the space to be heated (see Fig. 76). One large register at the top of the unit protects its top. The principle of air circulation is the same as in the pipeless furnace of Fig. 74. The products of combustion are conducted through a smokepipe, located in a crawl space below the floor, to a chimney.



(*Gilcor Products Corp.*)

FIG. 76. Floor furnace.

Floor furnaces are lower in first cost than any other type of warm-air furnace installation and may provide satisfactory heating of small homes in which all the rooms to be warmed are interconnected at all times. Two or more furnaces may be used for heating a home when the rooms are arranged in such a manner that they cannot be successfully heated with a single unit.

144. Current Research in Warm-Air Heating. Work in warm-air-heating research is now being continued in the Research Residence and in the Mechanical Engineering Laboratory at the University of Illinois, and it appears likely that changes may be made in both the gravity- and forced-circulation codes as a result of it. Figure 77 is a layout of the experimental warm-air and return-air ducts being used in the Research Residence during the winter of 1948–1949. It may be noted that both the warm-air trunk duct and the return-air trunk duct are

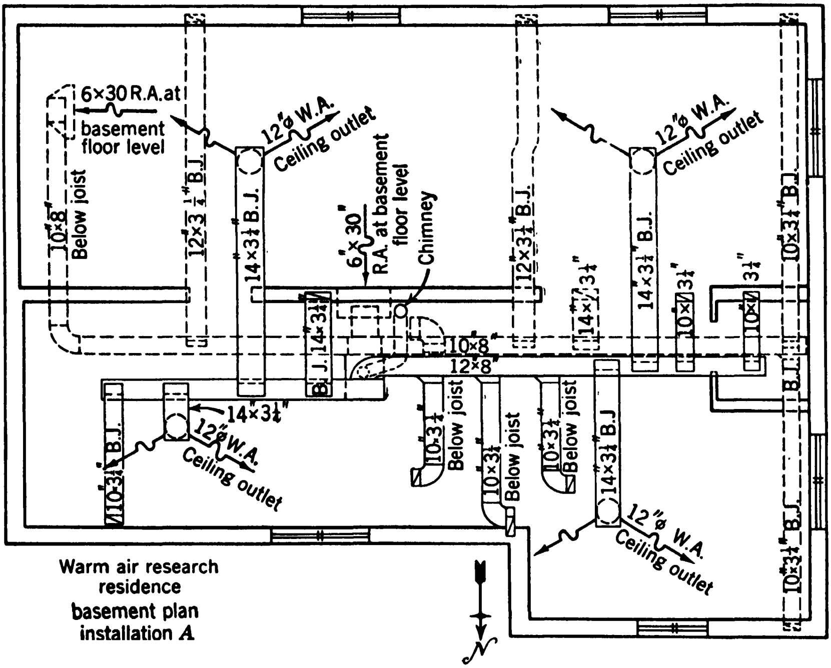


FIG. 77. Extended-plenum warm-air furnace system, Warm Air Heating Research Residence, Urbana, Illinois.

the same width and depth throughout their length. The system shown has performed satisfactorily in all types of weather, and it appears that this type of trunk duct may in time be adopted as standard by the National Warm Air Heating and Air Conditioning Association. Although a slightly greater amount of sheet metal is required in the fabrication of the extended-plenum type of trunk duct its use results in a considerable saving in labor and a reduction in cost. A view of a section of the warm-air trunk is shown in Fig. 78. Also shown in this picture is the bottom of a horizontal section of the return-air trunk, the vertical return-air plenum extending to the bottom of the furnace casing, the smoke pipe including the draft hood, and a new type of molded flue which is being used in place of a brick chimney.

A laboratory study is being made of many different types of trunk take-off fittings, other than those shown in Fig. 73. A new line of fittings is being developed which is designed for attachment to a hole cut in the side or in the top of a trunk duct of the extended-plenum type. Three side take-off fittings are shown in Fig. 78. It may be noted that the fittings shown terminate in the required stack dimensions rather than in the branch dimensions, thus eliminating one transition fitting which is required when the installation is made according to the code now in effect. Side and top take-off fittings are also being developed which will terminate in the proper round discharge opening to permit the use of lower-cost round pipe in the branch duct. Use of round pipe in the branch duct would necessitate the use of a more expensive boot, but if the length is unusually great the saving in duct cost might more than compensate for the increase in the cost of the boot.

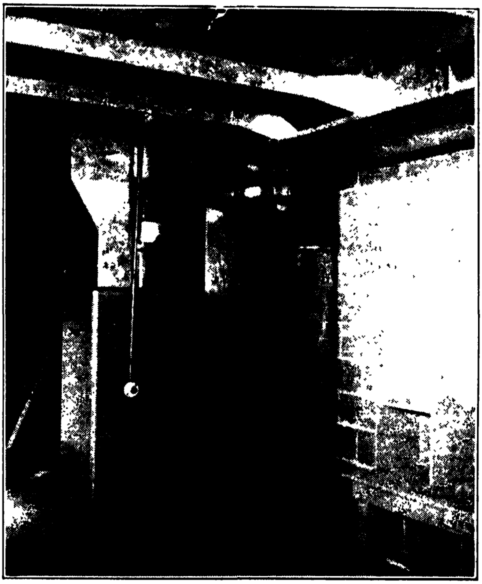


FIG. 78. Furnace as actually installed in the warm-air system shown in Fig. 77.

PROBLEMS

1. A gravity-flow warm-air steel furnace designed for hand-fired operation with coal as a fuel has a bonnet rating of 121,850 Btu per hr. The firepot has straight sides and a diameter of 22 in. inside its lining. Find: (a) the ratio of the heating

surface to the grate area, (*b*) the rating on the basis of register output, and (*c*) a possible rating in terms of square inches of leader cross-sectional area.

2. A cast-iron furnace suitable for gravity-flow warm-air heating has a firepot diameter of 26 in., a grate area of 2.38 sq ft, a heating surface of 45.9 sq ft, and a rating of 106,800 Btu per hr output at its bonnet. Find all possible ratings for both hand and automatic firing.

3. A gravity-flow warm-air furnace plant has a grate area of 2.38 sq ft and operates with an average equivalent register air temperature of 175 F. The bituminous coal used has a heating value of 11,230 Btu per lb as fired. Calculate the bonnet capacity of the furnace in Btu per hr. What percentage is this value of the manufacturer's rating of 116,800 Btu per hr? Base calculations on data from Fig. 55, and note that the grate areas of the two furnaces are different.

4. A second-floor room having heat losses of 9850 Btu per hr is to be served by a single gravity-flow warm-air furnace leader having a length of 8 ft and three elbows. Under the conditions of operation the leader is assumed to have a heat-carrying capacity of 167 Btu per hr per sq in. of cross-sectional area and the stack which serves it has a section equal to 75 per cent of the installed leader area. Find the size of required leader and stack by two methods, and compare the results.

5. A mechanical warm-air system is to supply 125,000 Btu per hr at the bonnet when operating with bituminous coal having a heating value of 10,450 Btu per lb as fired. If the equivalent register air temperature is to be 110 F, what grate area in square feet is necessary? How many pounds of coal will be burned per hour (see Fig. 56)?

6. Using heat losses 100 per cent greater than those stated in Fig. 66, find the sizes of the component parts of a gravity-flow warm-air furnace system for the building shown.

7. A structure having an occupied space of 16,000 cu ft is to be maintained at a dry-bulb temperature of 72 F with a corresponding wet-bulb temperature of 57 F when the outside-air temperature is 35 F and its relative humidity is 40 per cent. The barometric pressure is 29.92 in. of mercury. What weight of water vapor must be supplied per hour to maintain the required conditions when one change of air occurs each hour?

8. The furnace of problem 2 is to be converted to the use of gas, having a heating value of 1000 Btu per cu ft under standard conditions. Find the possible bonnet and register ratings of the furnace when used in a mechanical warm-air furnace plant.

9. A gas-fired mechanical warm-air furnace requires 180 cu ft per hr of fuel of 1000 Btu per cu ft to deliver 2020 cfm of air with a temperature rise of 70 F. The register output is stated to be 129,600 Btu per hr. (*a*) Find the overall efficiency of the furnace based on the probable output at its bonnet. (*b*) If the barometric pressure is 29.92 in. of mercury at what temperature are the 2020 cfm of air measured to give the calculated bonnet output?

10. The mechanical warm-air furnace system of Fig. 72 is to be installed in a residence of the same size and floor plans. However, the building construction and weather exposure are such that its heat losses are three times those shown in the example. Calculate the requirements of the plant in regard to all parts of the system.

CHAPTER 7

RADIATORS, CONVECTORS, AND BASEBOARD HEATING UNITS

145. Heat-Dispersal Units. When the fluids water and steam are used to convey heat from one location to another, surface areas warmed by the media mentioned are required to facilitate its release to the air and objects of spaces to be served. Certain assemblages of parts, which confine the fluids and which have external areas for heat emission, are termed either radiators or convectors.

146. Radiators. Such devices may be classified as direct, indirect, and direct-indirect units. Direct radiators are placed in the room to be heated; they transmit heat both by radiation and convection to objects heated. Indirect radiators are placed without the space to be heated; they deliver heat to air moving over them so that radiant heat is not delivered directly into the space to be heated by them. The heated air moves from indirect-radiator surfaces to the room to be heated. As a matter of fact, indirect radiators are a type of convector, although they are not generally classed as such. Direct-indirect radiators have sections which are exposed to view and also sections that are so enclosed as to make them indirect units. The housing, about the enclosed sections, has openings and damper arrangements at its bottom. Fresh air for ventilation purposes may be brought from the outside by a duct extending through the outside wall and then allowed to pass upward over the enclosed surfaces. The ventilating air is presumably warmed to a satisfactory temperature during its flow over the indirect areas. Provision may also be made for the circulation of room air over the indirect surfaces when the outside-air supply is shut off.

147. Direct Radiators. Many forms of direct radiators have been built and exist in present-day heating plants. Those radiators of commercial importance today are small-tube cast-iron, wall, cabinet, pipe-coil, finned pipe and baseboard units. Coils of various forms are built up with either steel or wrought-iron pipe and branch-tree headers. The other direct radiators mentioned are usually assemblages of cast-iron sections. Non-ferrous materials such as copper are little used in the construction of direct radiators alone.

148. Tubular Radiators. The manufacture of column and the larger tubular radiators has been discontinued. Because of its

TABLE 50

SMALL-TUBE DIRECT CAST-IRON RADIATORS—WIDTHS, HEIGHTS, AND EDR SURFACES*

Number of Sections	Length 1¾ In.† Per Sec	Three-Tube	Four-Tube			Five-Tube		Six-Tube		
		Heating Surface, Sq Ft	Heating Surface Sq Ft			Heating Surface, Sq Ft		Heating Surface, Sq Ft		
		25-In. Height 1.6 Sq Ft per Sec	19-In. Height 1.6 Sq Ft per Sec	22-In. Height 1.8 Sq Ft per Sec	25-In. Height 2.0 Sq Ft per Sec	22-In. Height 2.1 Sq Ft per Sec	25-In. Height 2.4 Sq Ft per Sec	19-In. Height 2.3 Sq Ft per Sec	25-In. Height 3.0 Sq Ft per Sec	32-In. Height 3.7 Sq Ft per Sec
2	3½	3.2	3.2	3.6	4.0	4.2	4.8	4.6	6.0	7.4
4	7	6.4	6.4	7.2	8.0	8.4	9.6	9.2	12.0	14.8
6	10½	9.6	9.6	10.8	12.0	12.6	14.4	13.8	18.0	22.2
8	14	12.8	12.8	14.4	16.0	16.8	19.2	18.4	24.0	29.6
10	17½	16.0	16.0	18.0	20.0	21.0	24.0	23.0	30.0	37.0
12	21	19.2	19.2	21.6	24.0	25.2	28.8	27.6	36.0	44.4
14	24½	22.4	22.4	25.2	28.0	29.4	33.6	32.2	42.0	51.8
16	28	25.6	25.6	28.8	32.0	33.6	38.4	36.8	48.0	59.2
18	31½	28.8	28.8	32.4	36.0	37.8	43.2	41.4	54.0	66.6
20	35	32.0	32.0	36.0	40.0	42.0	48.0	46.0	60.0	74.0
22	38½	35.2	35.2	39.6	44.0	46.2	52.8	50.6	66.0	81.4
24	42	38.4	38.4	43.2	48.0	50.4	57.6	55.2	72.0	88.8
26	45½	41.6	41.6	46.8	52.0	54.6	62.4	59.8	78.0	96.2
28	49	44.8	44.8	50.4	56.0	58.8	67.2	64.4	84.0	103.6
30	52½	48.0	48.0	54.0	60.0	63.0	72.0	69.0	90.0	111.0
32	56	51.2	51.2	57.6	64.0	67.2	76.8	73.6	96.0	118.4
34	59½	54.4	54.4	61.2	68.0	71.4	81.6	78.2	102.0	125.8
36	63	57.6	57.6	64.8	72.0	75.6	86.4	82.8	108.0	133.2
38	66½	60.8	60.8	68.4	76.0	79.8	91.2	87.4	114.0	140.6
40	70	64.0	64.0	72.0	80.0	84.0	96.0	92.0	120.0	148.0
42	73½	67.2	67.2	75.6	84.0	88.2	100.8	96.6	126.0	155.4
44	77	70.4	70.4	79.2	88.0	92.4	105.6	101.2	132.0	162.8
46	80½	73.6	73.6	82.8	92.0	96.6	110.4	105.8	138.0	170.2
48	84	76.8	76.8	86.4	96.0	100.8	115.2	110.4	144.0	177.6
50	87½	80.0	80.0	90.0	100.0	105.0	120.0	115.0	150.0	185.0
52	91	83.2	83.2	93.6	104.0	109.2	124.8	119.6	156.0	192.4
54	94½	86.4	86.4	97.2	108.0	113.4	129.6	124.2	162.0	199.8
56	98	89.6	89.6	100.8	112.0	117.6	134.4	128.8	168.0	207.2
58	101½	92.8	92.8	104.4	116.0	121.8	139.2	133.4	174.0	214.6
60	105	96.0	96.0	108.0	120.0	126.0	144.0	138.0	180.0	222.0
Distance from floor to center of top tapping, in.		23⅞	17⅞	20⅞	23⅞	20⅞	23⅞	17 11/16	23 11/16	30 11/16
Distance from floor to center of bottom tapping, in.		2½	2½	2½	2½	2½	2½	2½	2½	2½

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* National Radiator Company.

† Add ⅛ in. to length of all radiators for each bushing.

Width of sections, three-tube, 3⅜-in.; four-tube, 4 9/16-in.; five-tube, 5⅜-in.; six-tube, 7⅜-in. Width at feet is the same as width of section.

Tappings. Three, four, and five-tube radiators are tapped 1¼-in. at bottom both ends and 1-in. top, both ends. Six-tube tapped 1½-in. bottom both ends. 1¼-in. top both ends.

Can be furnished legless or with legs 4½-in. from floor to center of bottom tapping when ordered.

To determine overall height of three-tube, four-tube, or five-tube legless radiator, deduct 1⅜ in. from the standard heights. For six-tube deduct 1¼ in.

Radiators are not furnished in lengths exceeding 78 sections.

appearance, space requirements, and efficiency the small or junior-size tubular radiator, as shown by Fig. 79, is now the common unit. Commercial sections of this form of radiation are standardized as to heights, number of tubes, rated surface edr (equivalent direct radiation), linear length, and limits for the actual width of the sections. Table 50 includes data for tubular radiators, as built by one manufacturer, which conform to the accepted requirements.

Standardization dimensions placed on actual section widths range between the following minimum and maximum limits: 3-tube, $3\frac{1}{4}$ to $3\frac{1}{2}$; 4-tube, $4\frac{7}{16}$ to $4\frac{13}{16}$; 5-tube, $5\frac{5}{8}$ to $6\frac{5}{16}$; and 6-tube, $6\frac{13}{16}$ to 8 in. The radiators listed in Table 50 have widths between the foregoing values.

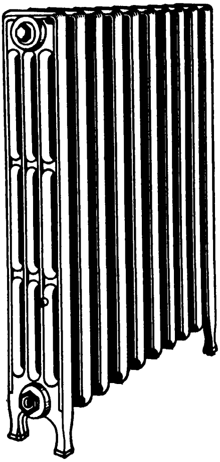


FIG. 79. Tubular radiator.

The required amount of surface in equivalent direct radiation, edr, Art. 150, in any radiator may be secured by fastening together a sufficient number of sections of the necessary height and width. The usual scheme of connecting the sections is by either malleable-iron screw or push nipples, Fig. 80. Details of the use of push nipples are shown in Fig. 81. Screw nipples require tapered threaded tappings in the radiator sections and do not necessitate the use of rods with nuts to hold them together.

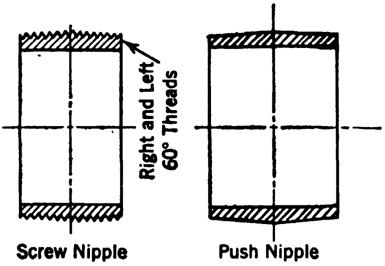


FIG. 80. Radiator nipples.

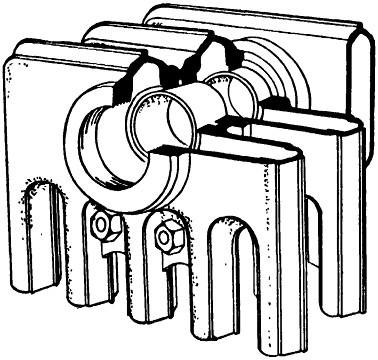


FIG. 81. Use of push nipples in a radiator.

The sections are further classified as loop and leg. The latter, of course, rest upon the floor, and the height of the radiator is based upon the nominal overall height of the leg sections. The loop sections are assembled between the leg sections and are listed as having the height of the leg sections with which they are used. Formerly radiators were also designated as steam and hot-water types. Steam-radiator sections are fastened together by nipples at their bottoms only. Such radiators cannot be used in hot-water heating because of the extreme difficulty of venting air from all the sections. Hot-water radiators have nipple connections between the sections at both the top and the bottom. Such radiators are absolutely necessary in hot-water heating systems where tubular units are used. As hot-water radiators may be used with any steam-heating system the manufacture of the steam-type unit has been discontinued.

Tubular radiators may be assembled without leg sections and hung on concealed brackets. Information relative to the actual lengths of loop sections or interior sections of the radiators listed in Table 50 is to be found in the footnotes of the table. Floor units may also be either assembled with sections having extra-high legs or used with pedestals. Old radiators with end sections having a bottom tapping for pipe connections lower than the other section bottom tappings are said to have low-drip hubs.

149. Standard Temperatures for Determining Radiator Performance. Definite temperature specifications are necessary in the standard conditions of performance of direct radiators. Steam units are supplied, when rated for heat transmission, with steam at 215 F and are tested in a room where the air temperature, at the breathing level, is 70 F, thus giving a steam-air-temperature difference of 145 F. In all tests the radiator is to stand in practically still air with the air movement over its surfaces due to natural convection currents. As the velocity of air flow has such a marked influence on the radiator heat output, other conditions being the same, it is necessary to have all radiators tested under comparable conditions as regards air movement. The so-called still-air conditions are the easiest to reproduce.

For hot-water radiators the temperatures are: water entering the radiator, 180; water leaving the radiator, 160; average temperature of the water in the radiator, 170; and room air, 70 F. Thus, for hot-water radiators, under standard conditions the average temperature difference between the water and the air is 100 F. The performance of hot-water radiators under test conditions is difficult to determine. Usually the performances of hot-water radiators are estimated from the heat emission per square foot obtained with the same radiators when using steam under standard conditions.

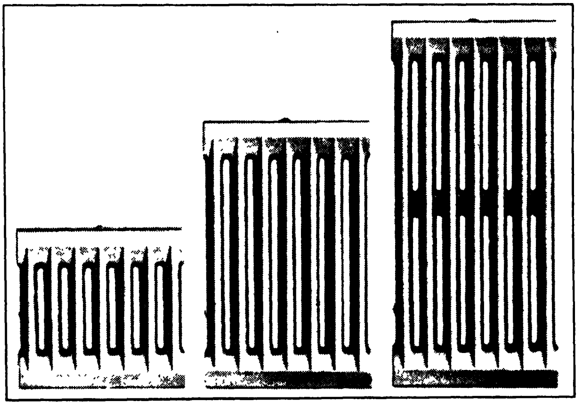
150. Heat Transmission of Direct Tubular Radiators. This class of unenclosed heat disseminator has its surface ratings, edr, sq ft of equivalent direct radiation, fixed on the basis of the emission of 240 Btu per hr per sq ft when operated with steam and 150 Btu per hr per sq ft when operated with hot water, the usual standard conditions of operation, for the respective media, being maintained in each case when the units are rated. The output of a radiator may also be expressed in terms of mbh (1000 Btu per hr).

Example. A 20-section, tubular direct cast-iron radiator gave off 24,000 Btu per hr, as determined from the weight of steam condensed. The steam was supplied at 215 F, and the room air temperature was 70 F with still-air conditions. Find the total surface in square feet at which the radiator is rated, also the rating in square feet per section.

Solution. The rated surface or equivalent direct radiation, is 24,000 ÷ 240 = 100 sq ft. The rated surface per section is 100 ÷ 20 = 5 sq ft of edr.

The radiator of the example when operated with hot water at an average temperature of 170 F and with the room air at 70 F will give out by the processes of radiation and convection 100 × 150 = 15,000 Btu per hr or 15 mbh.

151. Wall Radiators. Typical sections of one form of wall-type radiators are shown by Fig. 82. These sections may be assembled in a variety of ways by the use of suitable connecting nipples. The assembled units can be hung either on brackets along a wall, Fig. 83, or suspended under skylights or basement ceilings. The sections are easy to handle, and they compare favorably with pipe coils in the matter of heat output, Btu per hour per square foot of actual surface. Data pertinent to the physical dimensions and rated surface areas of the wall sections of Fig. 82 are given by Table 51.


5*A* 8*B* 11*B*

(*Kohler Co.*)

FIG. 82. Wall-radiator sections.

TABLE 51

DIMENSIONS AND HEATING SURFACES OF WALL RADIATOR SECTIONS*

Section Numbers	Height, In.	Length or Width, In.†	Thickness, In.	Distance Between Horizontal Top and Bottom Tappings, In.	Rated Heating Surface, Sq Ft EDR‡
5*A*	$12\frac{3}{4}$	$13\frac{5}{16}$	$2\frac{7}{8}$	$9\frac{3}{4}$	5
8*B*	$21\frac{3}{8}$	$13\frac{5}{16}$	$2\frac{7}{8}$	$18\frac{3}{8}$	8
11*B*	$28\frac{1}{2}$	$13\frac{5}{16}$	$2\frac{7}{8}$	$25\frac{1}{2}$	11

* Kohler Company.

† Add $\frac{1}{2}$ in. for each bushing and $\frac{7}{8}$ in. for each hexagonal nipple used in tappings.

‡ Based on 240 Btu per hr per sq ft for steam and 150 for hot water.

Under fixed operating conditions, wall-radiator sections mounted on walls give a greater output, Btu per hour per square foot, when they are assembled and placed on their hangers with the bars in a vertical rather than in a horizontal position. The placement of such sections in a horizontal position, beneath a ceiling, although favorable from the standpoint of heat transmission may lead to overheating of the upper portion of and the underheating of the lower part of the space so served.

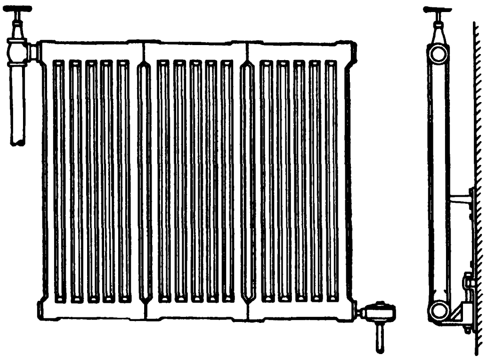
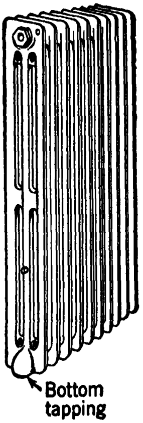


FIG. 83. Wall radiator with brackets.

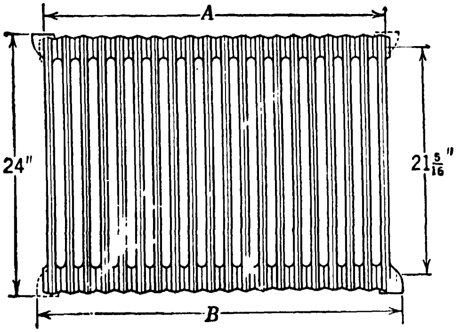
Loop or intermediate sections of small-size tubular radiators have a wide usage in wall mountings. Physical dimensions of loop sections of the radiators described in Art. 148 and their ratings, in square feet of equivalent direct radiation, edr, may be obtained from Table 50.

Another form of small-tube radiator for wall mounting is of the form indicated by Figs. 84 and 85. The sections, Fig. 85, which are used have a vertical height of 24 in. and a depth of $3\frac{1}{4}$ in. The sections may be assembled with horizontal-connection tappings at their tops and bottoms spaced a net distance apart equal to $21\frac{5}{16}$ in. as shown by Fig. 85. The rated surfaces of these radiators are based on the emission of 240 Btu per hr for steam and 150 Btu per hr per sq ft for hot-water operation. Ratings in terms of square feet of equivalent



(*United States Radiator Corp.*)

Fig. 84. Thintube wall radiator.



(*United States Radiator Corp.*)

Fig. 85. Dimensions of thintube wall radiator.

TABLE 52

Ratings and Dimensions of Thintube Wall Radiators*

Number of Sections	Heating Surface, Sq Ft EDR	*A*, In.	*B*, In.
2	3.2	$2\frac{5}{8}$	$4\frac{7}{8}$
4	6.4	$6\frac{1}{8}$	$8\frac{3}{8}$
6	9.6	$9\frac{5}{8}$	$11\frac{7}{8}$
8	12.8	$13\frac{1}{8}$	$15\frac{3}{8}$
10	16.0	$16\frac{5}{8}$	$18\frac{7}{8}$
12	19.2	$20\frac{1}{8}$	$22\frac{3}{8}$
14	22.4	$23\frac{5}{8}$	$25\frac{7}{8}$
16	25.6	$27\frac{1}{8}$	$29\frac{3}{8}$
18	28.8	$30\frac{5}{8}$	$32\frac{7}{8}$
20	32.0	$34\frac{1}{8}$	$36\frac{3}{8}$
22	35.2	$37\frac{5}{8}$	$39\frac{7}{8}$
24	38.4	$41\frac{1}{8}$	$43\frac{3}{8}$
26	41.6	$44\frac{5}{8}$	$46\frac{7}{8}$
28	44.8	$48\frac{1}{8}$	$50\frac{3}{8}$
30	48.0	$51\frac{5}{8}$	$53\frac{7}{8}$
32	51.2	$55\frac{1}{8}$	$57\frac{3}{8}$
34	54.4	$58\frac{5}{8}$	$60\frac{7}{8}$
36	57.6	$62\frac{1}{8}$	$64\frac{3}{8}$
38	60.8	$65\frac{5}{8}$	$67\frac{7}{8}$
40	64.0	$69\frac{1}{8}$	$71\frac{3}{8}$
42	67.2	$72\frac{5}{8}$	$74\frac{7}{8}$
44	70.4	$76\frac{1}{8}$	$78\frac{3}{8}$
46	73.6	$79\frac{5}{8}$	$81\frac{7}{8}$
48	76.8	$83\frac{1}{8}$	$85\frac{3}{8}$
50	80.0	$86\frac{5}{8}$	$88\frac{7}{8}$
Sq Ft per Linear In.	0.914		

* United States Radiator Corporation.

direct surface and physical dimensions indicated by *A* and *B* are included in Table 52.

152. Cabinet Radiators. These heat disseminators are built of cast sections fastened together with nipples and do not require any additional enclosure. The sections are so made, Fig. 86, that their edges butt together at the front forming a panel filled with the heating medium. Behind the front are cast tubes also filled with the heating

TABLE 53

HEATING SURFACES OF UNITED STATES RADIATOR CORPORATION CABINET RADIATOR

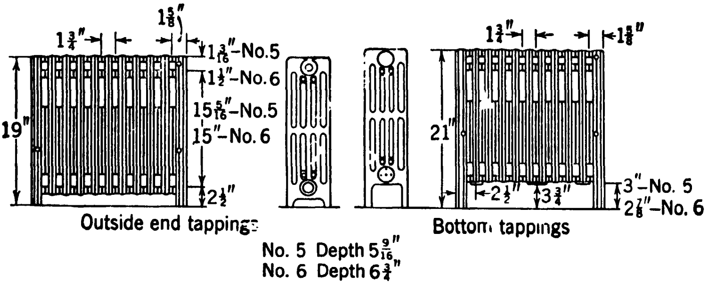
Number of Sections	Length, In.	Rating, Sq Ft EDR		Number of Sections	Length, In.	Rating, Sq Ft EDR	
		Number 5	Number 6			Number 5	Number 6
5	8½	9.0	11.5	29	50½	52.2	66.7
7	12	12.6	16.1	31	54	55.8	71.3
9	15½	16.2	20.7	33	57½	59.4	75.9
11	19	19.8	25.3	35	61	63.0	80.5
13	22½	23.4	29.9	37	64½	66.6	85.1
15	26	27.0	34.5	39	68	70.2	89.7
17	29½	30.6	39.1	41	71½	73.8	94.3
19	33	34.2	43.7	43	75	77.4	98.9
21	36½	37.8	48.3	45	78½	81.0	103.5
23	40	41.4	52.9	47	82	84.6	108.1
25	43½	45.0	57.5	49	85½	88.2	112.7
27	47	48.6	62.1				

Tappings.—Outside end tappings, number 5 tapped 1 in. top ends, 1¼ in. bottom ends. Number 6 tapped 1¼ in. top ends, 1½ in. bottom ends.

Bottom tappings are ¾ in. located in next-to-end sections unless otherwise specified. End-tapped radiators—add ½ in. to length for each bushing.

An allowance of 10 per cent extra is recommended when radiators are recessed.

medium. The air passageways between the tubes form flues which aid in promoting the circulation of air through the cabinet. The solid front face, exposed to the room, provides a surface which gives off heat by both radiation and convection. The heat transmission to air passing behind the front and over the tubes is by the process of convection. The air outlets may be either in the front at its top or in the top of the cabinet.



(United States Radiator Corp.)

FIG. 86. Cabinet radiator.

Information relative to the amounts of heating surface and ratings of one cabinet-type radiator are given in Table 53.

153. Pipe Coils. Radiators constructed of 1¼- and 1½-in. commercial wrought-iron and steel pipe made with suitable headers or manifolds, such as branch tees, have had considerable application in heating industrial plants. These coils are placed on the walls under and between window openings and under or adjacent to skylights in the roof. The coils may be of the miter or box type as shown by Fig. 87. The substitution of return bends at the pipe ends for manifolds or branch tees is not recommended. In pipe-coil radiators, ample provision must be made for expansion due to the rather long lengths of pipe that may enter into their construction.

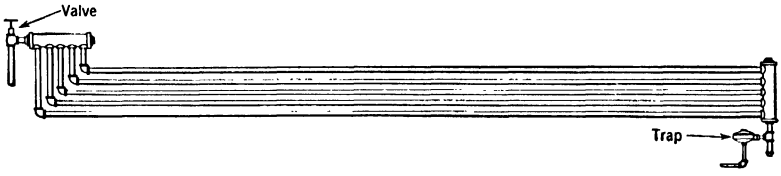


FIG. 87. Pipe-coil radiator.

The data of Table 54 indicate the thermal performance of horizontal pipe coils. Vertical positions of the pipes affect the operating characteristics of such units to some extent.

TABLE 54

Radiation Factors *R* for Direct Pipe-Coil Radiators with Steam

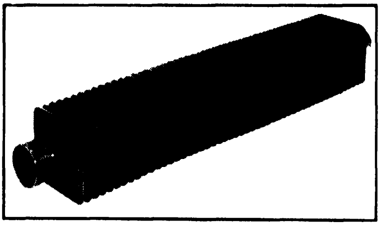
Steam temperature 215 F, room air 70 F, color black

Pipe-Coils Mounted on a Wall

Pipes in a horizontal position. British thermal units per hour per linear foot of coil

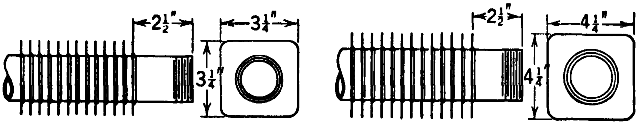
	Pipe Sizes, In.		
Rows of Pipes	1	1¼	1½
1	132	162	184
2	251	313	346
4	440	544	612
6	567	702	789
8	650	795	904
10	733	904	1015
12	814	1006	1134

Finned-pipe sections find usages where pipe coils might be employed. Figure 88 is indicative of the external appearance of finned steel pipe which is obtainable in 1¼- and 2-in. sizes in lengths which may be assembled along walls in the amounts necessary. External physical dimensions of the fins are shown in Fig. 89 for both the 1¼- and 2-in. pipe sizes. The fins are not bonded to the pipes, but they make firm contact with them to give good heat transmission through the areas of contact. Heating-surface ratings are 4¼ sq ft per linear foot of 1¼-in. pipe and 5¼ sq ft for the 2-in. size. The weights of pipe per linear foot are 4 lb for the 1¼-in. size and 6½ lb for the 2-in. sections. Brackets are used for wall mountings, and, when desirable, extended-metal grilles are used to cover installed units.



(*The Vulcan Radiator Co.*)

Fig. 88. Finned-tube radiator.



1¼· in. size

2· in. size

(*The Vulcan Radiator Co.*)

Fig. 89. Dimensions of finned-tube radiation.

154. Heat Dissemination from Direct Radiators. The heat given off by the radiator is received at its internal surfaces by contact with the conveying medium. This heat is carried by conduction through the metal walls of the radiator and is given off at the outer or radiating surfaces by the processes of radiation and convection. Resistance to heat transfer is encountered at both the inner and the outer surfaces of the sections. As the resistance to heat transfer at the inner surfaces is comparatively small and the outer-surface resistance to heat emission is greater to a varying extent the metal of the radiator section walls acquires a temperature practically equal to that of the medium in contact with it and the effect of the thickness of the thin section walls is negligible.

The amount of radiant heat given off by a direct radiator is usually less than one-half the total heat emitted. This heat is thrown out in straight lines to objects warmed, passing through the surrounding air without warming it appreciably. Certain radiator types have parts of their surfaces so placed as to enable them to emit more radiant heat than others. Figure 90 illustrates how exterior surfaces with more or less re-entrant angles have the emission of radiant heat partially blocked. The wider the radiator section, the greater the blocking effect. Because radiator sections are of irregular shape it is difficult but not impossible to apply theoretical coefficients for radiation and convection to them. Therefore the performance of radiators can best be determined by actual tests under operating conditions for each individual type of radiator. The unit heat transmission by radiation, i.e., the amount of heat given off per hour per square foot of surface, is independent of the extent and form of the surface, provided that re-entrant angles and re-entrant surfaces are not present. Radiant-heat emission is dependent upon the absolute temperature, the nature of the radiating surface, and the difference between the temperature of the hot surface and that of its surroundings. High temperatures of the surfaces are more conducive to an increase of heat transmission by radiation than are greater differences in temperature between the surface and its surroundings when the surface temperature is lower.

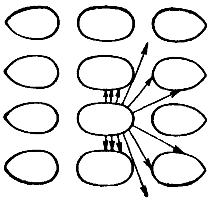


FIG. 90. Interception of radiant heat in a radiator.

Heat is transferred from radiator heating surfaces by convection as air moves over them. This method of heat transfer is not affected by the nature of the surfaces but is dependent upon their form and extent. Very important factors are the difference between the temperature of the surfaces and that of the surrounding air, and the velocity of air flow. Increase of velocity will produce an increase in the rate of heat transfer by convection, all other conditions remaining the same. The effect of increased air velocity is to decrease the surface film of stagnant air which offers resistance to the flow of heat. Rapid flow of water over the interior surfaces of steam and hot-water radiators also tends to produce increased transmission of heat.

155. Testing of Direct Radiators. The most trustworthy results will be obtained if radiators can be tested under actual operating conditions. Steam-condensation tests should be run in rooms where facilities are available for the maintenance of uniform experimental conditions. The radiator should stand in still air (movement due to natural convection currents only), and its output should not be affected by an environment which includes either hot or cold surfaces as either of these may change its output of radiant heat. Provisions must be made for the careful determination of the temperatures of the room air, steam, and condensate. Also required are accurate information relative to the steam pressure together with its quality and the weight of condensate during the test period.

The actual heat H_a, Btu per hour, given off by a direct radiator is

$$H_a = (h_f + xh_{fg} - h_{fc}) \frac{W_s}{D} \tag{80}$$

where h_f = enthalpy of the liquid of the steam supplied, Btu per lb.
x = quality of the steam, expressed decimally.
h_{fg} = enthalpy of evaporation of supply steam, Btu per lb.
h_{fc} = enthalpy of the liquid of the condensate, Btu per lb.
W_s = weight of steam condensed per test, lb.
D = duration of test period, hr.

The actual radiation factor R_a, Btu per hour per square foot of surface A, is

$$R_a = H_a \div A \tag{81}$$

When the temperature of the radiator steam t_s is 215 F and the room-air temperature t_r is 70 F the standard temperature difference of 145 F exists, and the actual value of H_a is equal to the value H_s for standard conditions. Likewise under the conditions mentioned R_a and R_s are equal. The expressions $H_a = R_aA$ and $H_s = R_sA$ are useful.

The American Society of Heating and Ventilating Engineers has codes[1] covering the testing of radiators and convectors. The ASHVE code method of expressing the heat emission H_s, Btu per hour, from a direct radiator operating under standard conditions is somewhat empirical, but it does give a satisfactory result. When the steam condensed is W, pounds per hour, the heat output per hour under standard conditions by the code is

$$H_s = 970W\left(\frac{215 - 70}{t_s - t_r)}\right)^{1.3} \tag{82}$$

156. The Location, Selection, and Installation of Direct Radiators. Whenever possible direct radiators should be placed below window openings, and their height should be such that they will not extend above the bottom of any window opening. Such locations are preferable to placements along interior walls or wall spaces adjacent to windows, for the radiators can more quickly warm the cold air entering at windows by infiltration. This arrangement also prevents or reduces cold drafts of air in the room especially at the floor level. Figures 91 and 92 indicate the circulation of air and its temperatures as determined by careful tests at the University of Illinois with direct radiators in two locations.[2]

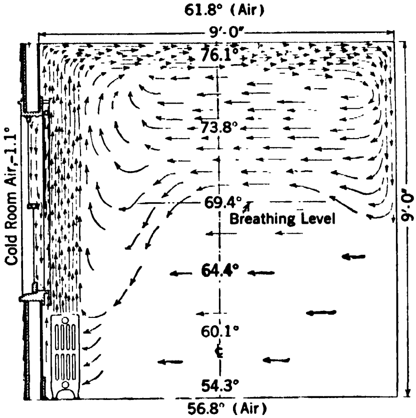


FIG. 91. Air circulation with unenclosed direct steam radiator.

The selection of radiators depends upon a number of items, such as the type of heating system, the wall and floor space available, the required amount of radiating surface, the allowable heights of the radiator sections, the type of radiation either available or most economical to use, and the use to which the heated space is put. The length of a cast-iron radiator has some effect on its performance in the matter of heat emission per square foot. The end sections are more effective in the emission of radiant heat, and also the air may move more freely over the ends, thereby increasing the heat transfer by convection. The decrease in heat transmission per rated square foot is quite appreciable when tubular radiators are increased in length from one to four sections. Beyond this point the reduction in output per unit of area is much less. As tubular radiators are seldom made up with less than four sections the effect of length of the radiator may be ignored.

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[1] "Code for Testing Radiators," *ASHVE Trans.*, Vol 33, 1927. "Standard Code For Testing and Rating Concealed Gravity Type Radiation (Steam) *ASHVE Trans.*," Vol. 37, 1931. (Hot Water) *ASHVE Trans.*, Vol. 39, 1933. See also *ASHVE Trans.*, Vol. 41, 1935, and Vol. 42, 1936.

[2] "Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields," by A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo, *University of Illinois Engineering Experiment Station Bulletin* 192.

Direct radiators should be centered below the window openings and placed to clear the adjacent wall by a distance $1\frac{1}{2}$ to $2\frac{1}{2}$ in. Before installation, radiators should have all possible accumulations of dirt and other foreign materials removed from their interiors. Cast-iron radiators longer than 40-section units are not desirable as their weight may make them hard to handle and also the sections are likely to be broken apart in handling. All radiators should be installed in a neat and workmanlike manner and so placed that water drainage from the outlet end is not hampered. Dividing the required amount of radiating surface among two or more units adds to the flexibility of operation but also somewhat increases the installation costs.

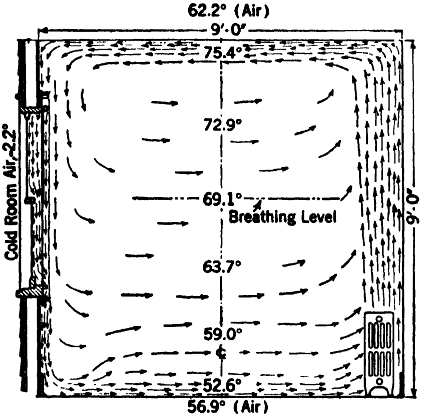

Fig. 92. Air circulation with unenclosed direct radiator at back wall.

157. Correction of Radiation Factors for Conditions Other Than Standard. Direct radiators in actual operation may have heating-media-air temperature differences more or less than 145 F. Therefore it is necessary to correct the factor for standard conditions to actual conditions which are not standard.

With tubular radiators the standard radiation factor is 240 Btu for steam and 150 Btu for hot-water service. The corrected factors for conditions other than standard are

$$R_a = 240\left(\frac{t_s - t_r}{215 - 70}\right)^{1.3} \text{ for steam} \qquad (83)$$

and

$$R_a = 150\left(\frac{t_w - t_r}{170 - 70}\right)^{1.3} \text{ for water} \tag{84}$$

where t_s = steam temperature, deg F.

t_w = average temperature of the water in the radiator, deg F.

t_r = room air temperature, deg F.

In either case the published rating in terms of equivalent direct radiation, edr, is multiplied by the corrected factor R_a to get the hourly heat output of the radiator.

Example. Find the hourly heat output of 100 sq ft of rated surface of direct cast-iron tubular radiators when operated with steam at 215 F and the room-air temperature at 65 F. Also find the hourly heat output when water enters the radiator at 190 F and leaves the radiator at 160 F.

Solution. For steam, $R_a = 240\left(\frac{215 - 65}{215 - 70}\right)^{1.3} = 250.7$ Btu per hr per sq ft

$$H_a = 250.7 \times 100 = 25{,}070 \text{ Btu per hr}$$

For hot water. The average water temperature equals 175 F.

$$R_a = 150\left(\frac{175 - 65}{170 - 70}\right)^{1.3} = 170 \text{ Btu per hr per sq ft}$$

$$H_a = 170 \times 100 = 17{,}000 \text{ Btu per hr}$$

The foregoing cases of the correction of direct radiator performance, for conditions other than standard, are easily made by the use of equations 83 and 84. When convectors are involved, Art. 167, the corrections are made in a similar manner using an exponent of 1.5 and the inlet air temperatures as 5 F less than those of the rooms. The data of Table 55 provide a means of rapidly making correction calculations for the stated conditions. Use of the data of Table 55 is made in the following case.

Example. Find the hourly heat output of a tubular radiator rated at 100 sq ft of equivalent direct surface when it operates with steam at 215 F and stands in a room having a breathing-line temperature of 65 F.

Solution. The prior calculation, using equation 83, gave an estimated output of 25,070 Btu per hr. When the data of Table 55 are used the calculated output becomes 100 × 240 ÷ 0.96 = 25,000 Btu per hr.

The operation of hot-water heating plants with direct radiators may involve the use of heated water where the operating conditions are not standard as the average temperature difference of the radiator

TABLE 55

Correction Factors for Direct Cast-Iron Radiators and Convectors*

Steam Pressure Approximately		Heating Medium Temperature F, Steam or Water	Factors for Direct Cast-Iron Radiators							Factors for Convectors						
			Room Temperature, F							Inlet Air Temperature, F						
Gage Vacuum, In. Hg	Psia		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Psig																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

* To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240, and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, the room temperature at 70 F for a radiator, and the inlet air temperature at 65 F for a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

water and the room air may be considerably more than 100 F. The water ratings of heating boilers are discussed in Chap. 8. The standard ratings of boilers as defined there must be modified in the proper selection of a unit when its actual conditions of operation are not as specified. Information relative to the heat emission and the boiler-rating factors are given in Table 56 for various average radiator-water

TABLE 56

Factors for Use with Direct Hot-Water Radiators*

Average Radiator Water Temperature, F	Radiation Factor at 70 F Air Temperature, Btu per Hr per Sq Ft Edr	Boiler-Rating Factor	Average Radiator Water Temperature, F	Radiation Factor at 70 F Air Temperature, Btu per Hr per Sq Ft Edr	Boiler-Rating Factor
150	110	0.733	195	200	1.333
160	130	0.866	200	210	1.400
170	150	1.000	210	230	1.533
180	170	1.133	215	240	1.600
190	190	1.266			

* Burnham Boiler Corporation.

temperatures when the breathing-line air temperature of the room is 70 F. When the average water temperature in a radiator is more or less than 170 F a boiler may be selected with a net load rating equal to the required amount of installed radiation multiplied by the proper rating factor as given in Table 56 for the average radiator-water temperature.

158. Heating Effect of Direct Radiators. The condensation of steam per hour in a radiator is not a complete measure of its performance as certain radiators will do more than others in maintaining a smaller temperature differential between the floor and the ceiling of a room. This may be accomplished by the use of the same or smaller amounts of steam condensed per hour. The larger portion of the air-temperature gradient between the floor and ceiling of a room occurs between the floor and breathing line when the heating is done with direct radiators. Air temperatures at a level 30 in. above the floor are more indicative of the heating effect of a radiator than those at the breathing line. Heating effect is defined as the ratio of the heat output of a radiator held by the air in the occupied zone to the total heat output of the radiator. More nearly uniform air temperatures from floor to breathing line are preferable from the standpoint of comfort, and smaller temperature differentials from the breathing line

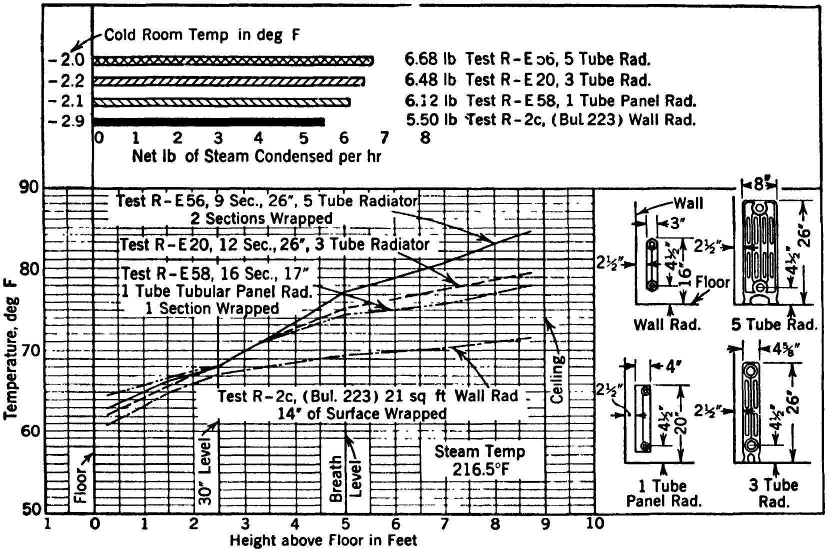

FIG. 93. Room-air temperature gradients and steam-condensation rates for four cast-iron direct radiators.

to the ceiling are beneficial in reducing the heat losses from the room. Long, low, thin, direct radiators maintain a lower air-temperature differential from floor to ceiling than high and wide radiators when both are operated with the same breathing-line air temperature and the same steam temperature. Data[3] relative to the heating effects of high tubular and low narrow radiators are shown by the curves of Fig. 93.

159. Effects of Radiator Finishes on Direct-Radiator Performance. Surface finishes change the amount of heat given out from a unit by radiation but do not change the amount of heat given off by convection. A surface effect is involved; therefore the last coat of the finish determines the surface nature, and the undercoats of finish have no appreciable bearing in the matter. Finely ground pigments have for the most part about the same radiation properties as black cast iron and usually produce little change in radiator performance when incorporated in radiator paints. Bright surfaces transmit less heat by radiation than dull surfaces. Bronze metallic paints reduce the transmission factors of direct radiators. Table 57 gives the results obtained by tests to determine the effect of radiator finishes on two types of direct cast-iron radiators.

TABLE 57

EFFECT OF SURFACE FINISHES ON DIRECT CAST-IRON RADIATORS

Three-column, 6-section, 32-in. cast-iron radiators*

Radiator Number	Finish	Area, Sq Ft	Coefficient of Heat Transmission	Relative Heating Value Per Cent
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat aluminum bronze	27	1.60	90.8
3	Gray paint, dipped	27	1.78	101.1
4	One coat dull black paint	27	1.76	100.0

Four-tube, 10-section, 26-in. cast-iron radiators†

Finish	Relative Heat Emission	Finish	Relative Heat Emission
Bare radiator	100.0	Flat brown paint	104.8
Aluminum bronze	93.7	Flat cream paint	104.0
Gold bronze	92.6	White gloss enamel	102.2

* "Comparative Tests of Radiator Finishes," by W. H. Severns, *ASHVE Trans.*, Vol. 33, 1927.

† "Experiments on the Effect of Surface Paints on Radiator Performance," by C. H. Fessenden and Axel Marin, *ASHVE Trans.*, Vol 35, 1929.

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[3] "Steam Condensation an Inverse Index of Heating Effect," by A. P. Kratz and M. K. Fahnestock, *ASHVE Trans.*, Vol. 37, 1931.

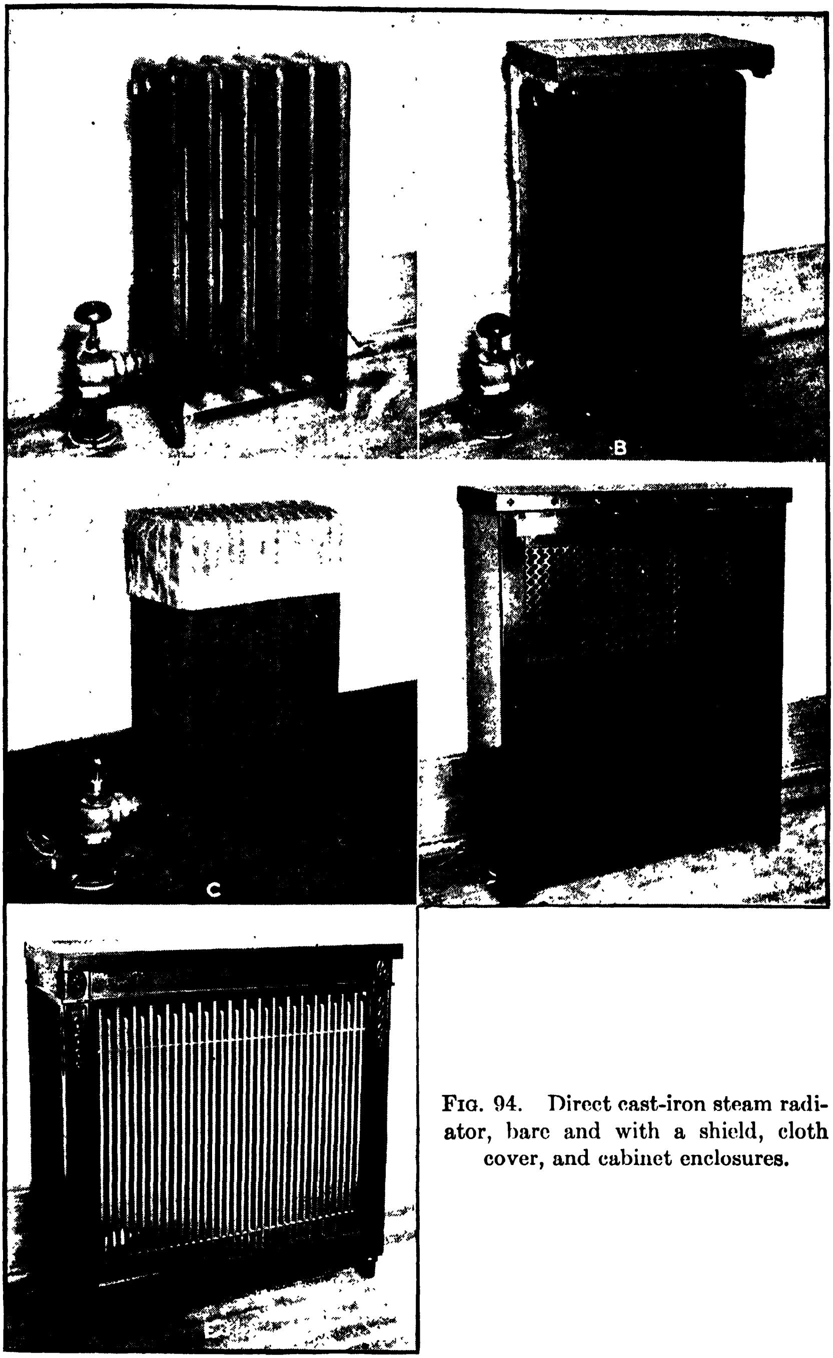

FIG. 94. Direct cast-iron steam radiator, bare and with a shield, cloth cover, and cabinet enclosures.

The effect of the surface finish is greater with any type of radiator which has a large proportion of its total surface effective for the emission of radiant heat. Four-column radiators will be less affected than single-column radiators of the same height. Greater effects may be expected with pipe-coils and wall sections than with tubular radiators 4, 5, or 6 tubes wide, all other conditions being the same as to height and methods of operation.

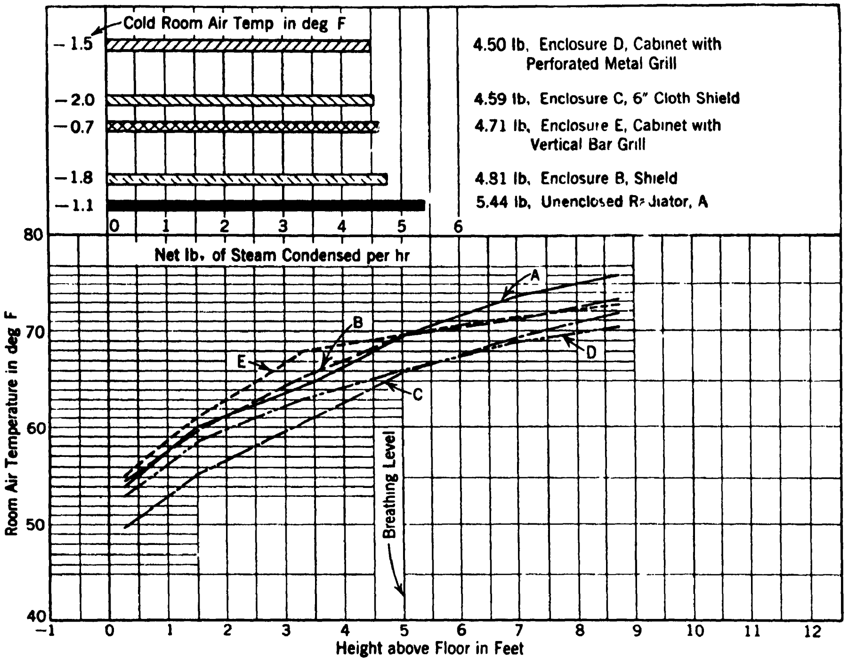

FIG. 95. Effect of a shield, cloth cover, and cabinet enclosures on the performance of a direct cast-iron steam radiator.

160. Effects of Radiator Enclosures with Direct Radiators. Any enclosure placed about a direct steam radiator will reduce its steam condensation per hour unless the enclosure is extended to a level well above the top of the radiator. In any enclosure the air flow at both the bottom inlet and the top outlet should not be restricted. The final heating effect produced by the enclosed unit depends greatly upon the design of the enclosure of a radiator. Some enclosures placed over an installed radiator will produce better heating effects in a room, although the steam required per hour is less; other enclosures produce no effect other than to reduce the consumption of steam; and finally poorly designed cabinets and covers will so reduce the heating effect of the radiator unit that additional units must be installed if the space is to be properly heated. The effects[4] of various radiator enclosures, shields, and covers, Fig. 94, on room-air temperature gradients and steam condensation are shown by Fig. 95 for a single radiator so equipped. The distribution of heated air in a room having a good enclosure fitted to a direct steam radiator is indicated by Fig. 96.

The effect of radiator enclosures of the cabinet type is to change direct radiators into some form of a convector. Direct radiators are sometimes installed in recesses in walls with or without decorative panels in front of them. Their performance is affected by the height of the radiator used, the nature of the recess in the wall, and the free area openings of the air inlets and outlets and their locations. In general the effect of placing the radiator in a recess is to reduce the heat output of a given unit by about 10 per cent. This statement does not apply to properly designed convectors which are described in Art. 163.

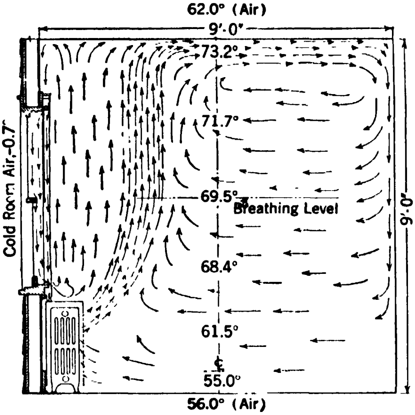

FIG. 96. Air circulation in a room with a direct radiator enclosure.

161. Weights and Volumes of Direct Radiators. In hot-water heating it may be important to estimate the weight of water-filled radiators supported by wall brackets. Furthermore, it is necessary

TABLE 58

WEIGHTS AND INTERNAL VOLUMES OF DIRECT RADIATORS PER SQUARE FOOT OF RATED SURFACE

Radiator	Weight per Square Foot, Lb	Volume per Square Foot, Pints	Radiator	Weight per Square Foot, Lb	Volume per Square Foot, Pints
Column	7	1.3	Wall sections	6	1.7
Tubular, large	5.8	0.8	Pipe coils 1¼-in. and		
small	4.2	0.5	1½-in. pipe	5.3	1.6

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[4] "Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields," by A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo, *University of Illinois Engineering Station Bulletin* 192.

to know the volume of water contained in the radiators in order to estimate the size of an expansion tank, when used, for a hot-water heating system. The weights of radiator sections and their internal volumes vary with the type of section and also somewhat with the different manufacturers. The data given in Table 58 are more or less average values which are adequate for most calculations.

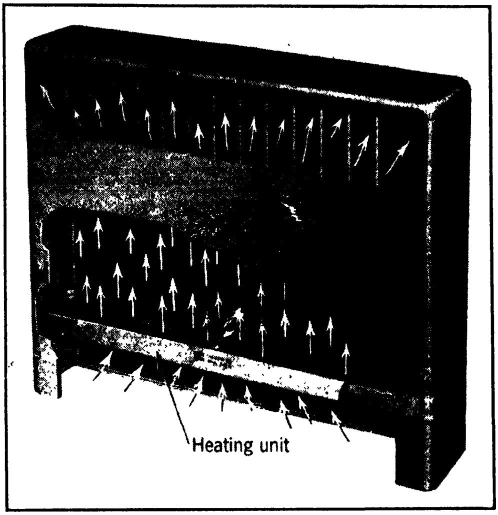
162. Radiator Tappings and Bushings. Some of the older styles of cast-iron radiator sections have tapped openings in their end sections, at the level of the bottom row of connecting nipples, as large as 2-in. pipe size. Other radiators are tapped for $1\frac{1}{2}$-in. pipe connections. The modern tendency with radiators having tubes of small size is to provide them with pipe openings as small as $1\frac{1}{4}$- to 1-in. size. The data of Table 59 designate the common or bushed-opening pipe-connection sizes for various amounts of radiation employed in the different kinds of heating plants. All air-valve openings are tapped for $\frac{1}{8}$-in. pipe.

TABLE 59

DIRECT CAST-IRON RADIATOR BUSHING PIPE SIZES

Radiator Heating Surface, Sq Ft	One-Pipe Steam	Two-Pipe Steam		Vapor		Hot-Water	
		Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
0 to 24 inc.	1						
Above 24 to 60 inc.	$1\frac{1}{4}$						
Above 60	$1\frac{1}{2}$						
0 to 48 inc.	...	1	$\frac{3}{4}$				
Above 48 to 96 inc.	...	$1\frac{1}{4}$	1				
Above 96	...	$1\frac{1}{2}$	$1\frac{1}{4}$				
0 to 30 inc.	...	...	.	$\frac{1}{2}$	$\frac{1}{2}$	1	1
Above 30 to 60 inc.	...	...	...	$\frac{3}{4}$	$\frac{1}{2}$	...	...
Above 60 to 125 inc.	...	...	...	1	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$
Above 125	...	...	...	$1\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
0 to 40 inc.	...	...	...	...	...	1	1
Above 40 to 72 inc.	...	...	...	...	...	$1\frac{1}{4}$	$1\frac{1}{4}$
Above 72	...	...	...	...	...	$1\frac{1}{2}$	$1\frac{1}{2}$

163. Convectors. The term convector covers a great variety of concealed heating units constructed in many forms of both ferrous and non-ferrous materials. One form of such equipment functions with a heating element placed at a low level within an enclosure as shown



(*Young Radiator Co.*)

FIG. 97. Cabinet convector with finned-tube heating element.

by Fig. 97. The devices considered here differ from either direct radiators fitted with enclosures, as shown by Fig. 94, or direct radiators recessed into walls. Because of the chimney action produced by the effects of stack height, Fig. 98, and the differences in the temperatures of air within and without the enclosure, cool air from near a room floor is circulated through the heating element of a convector. The air thus warmed is discharged through openings either in the top of or through a grille placed near to the top of the assembled unit.

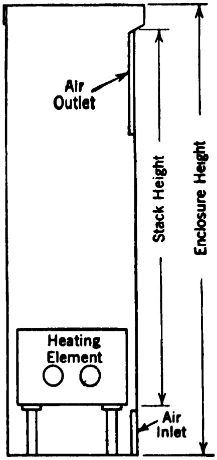


FIG. 98. Stack and enclosure heights of a convector.

Three general types of convector enclosures are available; these are cabinet, partly concealed, and totally concealed units. Thus convectors may be installed under window or glass areas either as free-standing floor cabinet units, Fig. 97; assemblages enclosed in wall-mounted cabinets; recess-types partially set into the wall, Fig. 102; recess types fully set into the wall with a front panel flush with its surface, Fig. 102; and finally those fully concealed arrangements with their heating units placed within wall spaces behind plaster coverings with the inlet and air outlet visible as in Fig. 99. The partially concealed or semi-recessed types are generally installed under window openings with a portion of the cabinet within the wall and part of the cabinet extending into the room. The air outlets may be located either in the window stool above the unit or near to the top of its enclosure front to discharge heated air horizontally into the room.

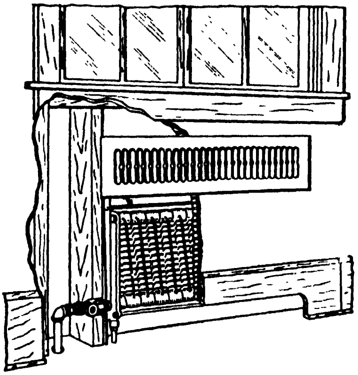
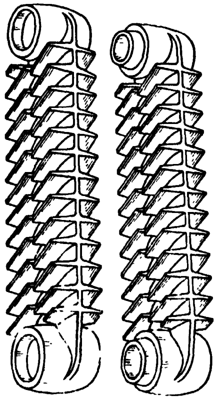
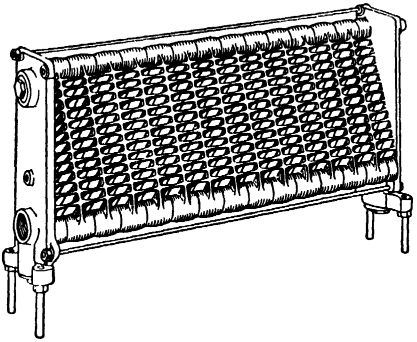


Fig. 99. Cast-iron convector with wall concealment.



(*National Radiator Co.*)
Fig. 100. Cast-iron Aero convector sections.

The heating elements of convectors may be classified by position as vertical, horizontal, and inclined. Small-tube cast-iron sectional radiators lend themselves as vertical heating elements in both free-standing cabinet and wall-enclosed convectors. Other sectional heating units of cast iron, Figs. 100 and 101, have finned projections which are integral with the core of the element and which add to its air-heating surfaces. Non-ferrous materials used for the construction of heating-element cores include copper, brass, and aluminum. With non-ferrous materials one common form of heating element design involves inlet- and outlet-connection headers joined by tubes of either brass or copper which have attached finned extensions of either sheet copper, brass, or aluminum as shown by



(*National Radiator Co.*)
Fig. 101. Aero convector, series B, 5½ in., 15 section, single unit.

Fig. 97. Where extended air-heating surfaces are formed from sheet materials of either copper or brass, bonding between tubes and fins may be secured by soldering. Good contact between non-ferrous tubes and fins is often secured by heavy pressure exerted upon fin and tube parts during the assembly of the core of the heater unit.

The rate of air flow through a convector has much to do with the output obtained. Furthermore this flow of air must be through the heating element as it should not by-pass the surfaces where it may be properly warmed. The design of a convector-enclosure air inlet and also its air outlet affect its performance, and consideration should be given to these items. The capacity of a convector is dependent upon the height of the stack for any given set of operating and design conditions. The greater the stack height the greater is the heat output per hour for a given type of heating unit. However, the high-outlet convectors tend to give poorer heating effects owing to increased tendency for air stratification in a room. The general tendency is to avoid the use of convectors which cannot be placed beneath the usual window opening.

164. Concealed and Semi-Recessed Convector Fronts. Removable decorated metal panels are provided for some concealed and also semi-recessed convectors, which make the heating elements entirely accessible. Other convectors have their stack-front panels unfinished, and these serve as a plaster base; or lath and plaster may be placed in front of the metal of the stack, the top and the bottom outlet and inlet grilles being removable so that access may be had to the heating element or unit.

165. Control of Convectors. The heat output of a convector may be regulated by either of two methods. The flow of the heating medium may be controlled either by a valve in the heating-medium supply connection as in direct radiators or by a damper at the air outlet of the unit. Closure of the air damper stops air circulation and the unit becomes inactive. A few convectors are provided with both forms of control.

166. Uses of Convectors. Heat-dispersal units in many heating plants are convectors built with provision for separate heating-medium inlet and outlet connections in vapor, vacuum, and other two-pipe steam systems and for use with either gravity-flow or forced-circulation hot-water installations. Not every convector as constructed is an all-purpose unit. However, most manufacturers are able to supply designs suitable for one-pipe steam-heating systems and also units for hot-water heating when their regular convectors for two-pipe steam usage are not adaptable to hot-water heating.

167. Ratings of Convectors. As for direct radiators, Art. 155, convectors may be rated on the basis of steam-condensation tests. When the proper air temperatures and steam data have been secured the actual heat output of a convector unit H_{acv} Btu per hour, can be computed by use of an expression identical with equation 80. Use can also be made of an equation similar to number 81 to find a radiation factor R_{acv} in Btu per hour per square foot of rated surface.

Ratings for steam operation are standardized for the conditions of 215 F steam temperature within the heating element and air entering the convector enclosure at 65 F. By the ASHVE Code the heat output of a steam convector operating under standard conditions is

$$H_{scv} = h_{fg}W\left[\frac{215 - 65}{t_s - t_i}\right]^{1.5} \tag{85}$$

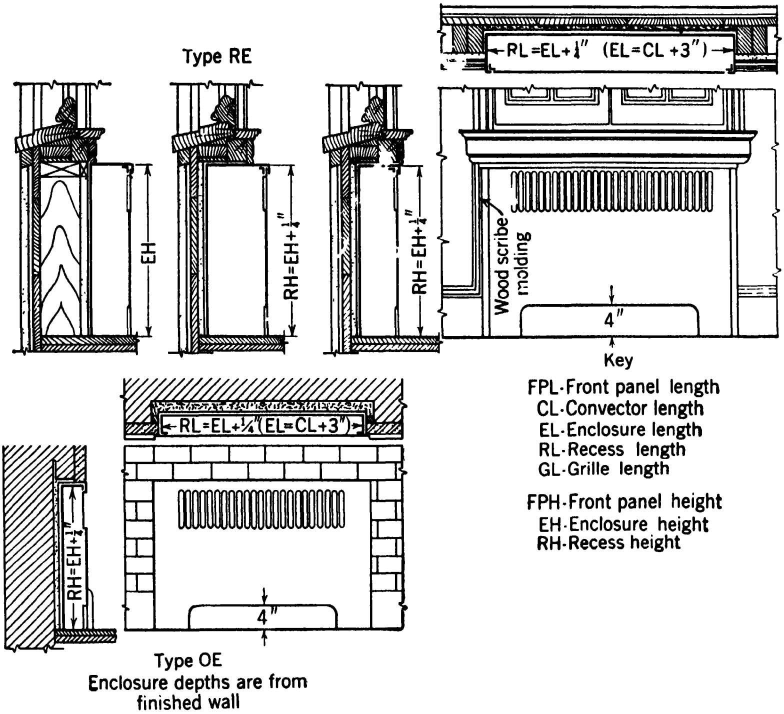
where h_{fg} = enthalpy of evaporation at steam pressure, Btu per lb.
W = steam condensed, lb per hr.
t_s = steam temperature, deg F.
t_i = inlet-air temperature, deg F.

In the foregoing equation t_i is not equal to the room temperature t_r, and also the exponent applied to the temperature-difference ratio is an average value of 1.5 for convectors and not the value of 1.3 as used for direct radiators. The outputs may be given as the maximum and minimum square feet of equivalent direct radiation, edr, of direct radiators or as the total Btu output per hour. Frequently an extra allowance is included in the rating for heating effect. This is an extra percentage added to the rating obtained on the basis of the condensing capacity of the convector and is claimed, by the maker, because of the better heat distribution possible in the occupied zone when a convector is used. Ratings as given should be studied critically to determine whether the allowance added for heating effect is excessive.

Views illustrative of finned cast-iron heating sections and their assemblage into a heating element for a convector are shown by Figs. 100 and 101. The enclosure arrangements of cabinet, partially recessed, and fully recessed flush-front panel-coverage, for such units, are indicated by Fig. 102. The manufacturer's ratings for different section heights, depths, and lengths of section assemblages are given by Table 60, page 228.

168. Heating Performance of Convectors for Conditions Other Than Standard. Many convectors cannot be operated with steam at 215 F and the inlet air temperature at 65 F. When the conditions of operation are other than standard it is necessary to estimate from the rating data the performance to be expected under the actual conditions of operation. The corrected radiation factor is

$$R_{cv}' = 240 \left[\frac{t_m - t_i}{215 - 65} \right]^{1.5} \tag{86}$$



(*National Radiator Co.*)

FIG. 102. Dimensions of enclosures for Aero convectors.

where R_{cv}' = the actual heat ouput per sq ft of rated area, A in edr, Btu per hr.

t_m = average temperature of the heating medium used, deg F.

t_i = inlet temperature of the air, deg F.

$H_{cv}' = A \times R_{cv}'$ = total heat output of the convector under actual conditions, Btu per hour. The value of the exponent of equation 86 as determined by tests ranges from 1.27 to 2.07. When the ratings are given in total Btu per hour output the corrected value for conditions other than standard becomes

$$H_{cv}' = H_{cv} \left[\frac{t_m - t_i}{215 - 65} \right]^{1.5} \tag{87}$$

TABLE 60

AERO CONVECTOR STEAM OR WATER RATINGS WHEN USED WITH FRONT OR TOP-OUTLET ENCLOSURES SQUARE FEET, EDR*

$3\frac{3}{4}$-In. Depth Enclosures, $4\frac{3}{8}$-In. Depth Enclosures, $5\frac{7}{8}$-In. Depth Enclosures

Lengths			Overall Heights of Enclosures						Overall Heights RE and OE Enclosures, In.													
Nominal Convector†	Complete Enclosure‡	Front Panel‡	Front Panels Should Be Specified 1-In. Higher Than Corresponding Enclosures						All Front Panels Should Be Specified 1 In. Higher Than Corresponding Enclosures§													
Inches			Convector Number	20	23	26	29	32	Convector Number	20	23	26	29	32	Convector Number	20	Convector Number	21	23	26	29	32
				3B Convectors‖						4B Convectors						5B		5C Convectors				
16	19	21	3B16	10	11	11	12	12	4B16	10	11	12	12	13	5B16	11	5C16	13	14	15	16	16
18	21	23	3B18	11	12	13	13	13	4B18	11	13	13	14	14	5B18	13	5C18	14	15	17	18	18
20	23	25	3B20	12	13	14	14	15	4B20	13	14	15	15	16	5B20	14	5C20	16	17	19	20	20
22	25	27	3B22	14	15	15	16	16	4B22	14	15	16	17	17	5B22	16	5C22	18	19	20	22	22
24	27	29	3B24	15	16	17	17	18	4B24	15	17	18	18	19	5B24	17	5C24	19	20	22	24	24
26	29	31	3P26	16	17	18	19	19	4B26	16	18	19	20	20	5B26	19	5C26	21	22	24	26	26
28	31	33	3B28	17	18	19	20	21	4B28	18	19	21	21	22	5B28	20	5C28	22	24	26	28	28
30	33	35	3B30	18	20	21	22	22	4B30	19	21	22	23	23	5B30	21	5C30	24	26	28	30	31
32	35	37	3B32	20	21	22	23	24	4B32	20	22	24	24	25	5B32	23	5C32	26	27	30	32	33
34	37	39	3B34	21	22	24	25	25	4B34	21	24	25	26	27	5B34	24	5C34	27	29	32	33	35
36	39	41	3B36	22	24	25	26	27	4B36	23	25	27	27	28	5B36	26	5C36	29	31	33	35	37
38	41	43	3B38	23	25	26	27	28	4B38	24	26	28	29	30	5B38	27	5C38	30	32	35	37	39

40	43	45	3B40	25	26	28	29	30	4B40	25	28	30	30	31	5B40	28	5C40	32	34	37	39	41
42	45	47	3B42	26	28	29	30	31	4B42	26	29	31	32	33	5B42	30	5C42	34	36	39	41	43
44	47	49	3B44	27	29	31	32	33	4B44	28	31	33	33	34	5B44	31	5C44	35	37	41	43	45
46	49	51	3B46	28	30	32	33	34	4B46	29	32	34	35	36	5B46	33	5C46	37	39	43	45	47
48	51	53	3B48	29	32	33	35	36	4B48	30	33	35	36	38	5B48	34	5C48	38	41	45	47	49
50	53	55	3B50	31	33	35	36	37	4B50	31	35	37	38	39	5B50	36	5C50	40	43	46	49	51
52	55	57	3B52	32	34	36	38	39	4B52	33	36	38	39	41	5B52	37	5C52	42	44	48	51	53
54	57	59	3B54	33	36	37	39	40	4B54	34	38	40	41	42	5B54	38	5C54	43	46	50	53	55
56	59	61	3B56	34	37	39	40	42	4B56	35	39	41	42	44	5B56	40	5C56	45	48	52	55	57
58	61	63	3B58	36	38	40	42	43	4B58	36	40	43	44	45	5B58	41	5C58	46	49	54	57	59
60	63	65	3B60	37	39	42	43	45	4B60	38	42	44	45	47	5B60	43	5C60	48	51	56	59	61
62	65	67	3B62	38	41	43	45	46	4B62	39	43	46	47	48	5B62	44	5C62	50	53	58	61	63
64	67	69	3B64	39	42	44	46	48	4B64	40	44	47	48	50	5B64	46	5C64	51	54	59	63	65
66	69	71	3B66	41	43	46	48	49	4B66	41	46	49	50	52	5B66	47	5C66	53	56	61	65	67
68	71	73	3B68	42	45	47	49	51	4B68	42	47	50	51	53	5B68	48	5C68	54	58	63	67	69
70	73	75	3B70	43	46	49	51	52	4B70	44	49	52	53	55	5B70	[illegible]	5C70	56	60	65	69	71
72	75	77	3B72	44	47	50	52	53	4B72	45	50	53	54	56	5B72	51	5C72	58	61	67	71	73
74	77	79	3B74	45	49	51	53	55	4B74	46	51	55	56	58	5B74	53	5C74	59	63	69	73	75
76	79	81	3B76	47	50	53	55	56	4B76	47	53	56	57	59	5B76	54	5C76	61	65	71	75	77
78	81	83	3B78	48	51	54	56	58	4B78	49	54	58	59	61	5B78	55	5C78	62	66	72	77	79
80	83	85	3B80	49	53	55	58	59	4B80	50	56	59	60	63	5B80	57	5C80	64	68	74	79	81
82	85	87	5B82	50	54	57	59	61	4B82	51	57	60	62	64	5B82	58	5C82	66	70	76	81	83

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* National Radiator Company.

† For one-pipe steam, 3B, 4B, 5B, and 5C convectors are 1½-in. longer than shown in this column.

‡ Length when used with piping connections 1, 2, 3, or 4; add 7-in. when used with 5, 6, 7, 8, 9, 10, 11, or 12. Length with 13 or 14 is variable.

Specify 6-in. legs for 5C with 21-in. high enclosure or recess. RE and OE enclosures are furnished with or without removable angle at front of liner. Either angle or legs must be temporarily removed to install 5C in 21-in. high enclosures. Place convector to rear to avoid interference with angle.

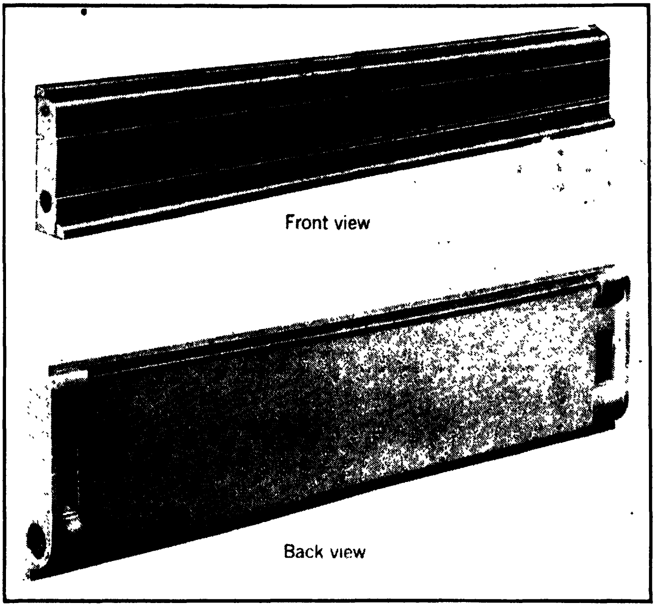
§ All front panels should overlap recess 1-in. at top and each end.

‖ 3B convector is used only where depth of 4B convector is too deep for recess.

To obtain ratings with wall-hung enclosures add 6-in. to the height, and use the above table. Ratings with top-outlet enclosures may be increased 10 per cent in 20-in. EH, 5 per cent in 25-in. EH, and 2 per cent in 26-in. EH. Above 26-in. EH no increase.

where H_{cv}' = total heat output for actual conditions, Btu per hr.
H_{cv} = total heat output for standard conditions, Btu per hr.
t_m = average temperature of the heating medium, deg F.
t_i = inlet-air temperature, deg F.

169. Baseboard Heating Units. Prior discussions of radiator performances have indicated that better heating effects can be obtained and smaller air-temperature differences maintained between floor and ceiling levels with low, narrow, and long direct radiators than are possible with taller and wider units. Other desirable features with any heating system include inconspicuous locations of heat-dispersal units and their non-interference with room decorations, draperies, and furniture as well as uniformity of heat release about the rooms. Developments have been made with heating units which replace the usual baseboard placed in a room just above the floor level and which have most of the desirable operating characteristics previously listed.

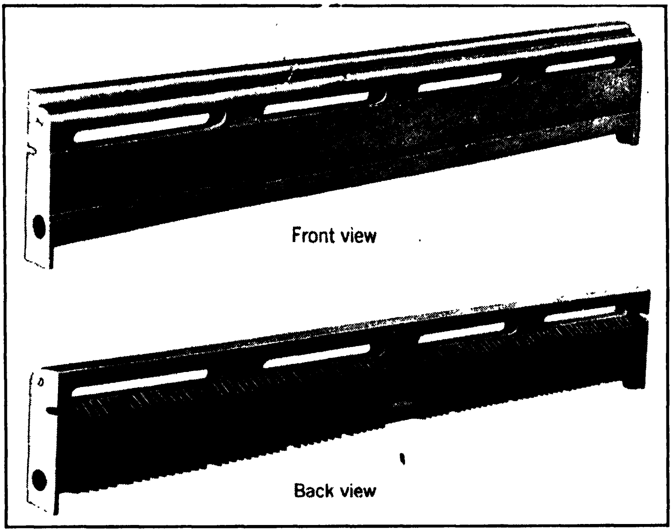


(Burnham Corp., Boiler Division)

FIG. 103. Type-R radiant baseboard.

Baseboard units may be divided into two general groups, which are direct-radiant elements and those which primarily are convectors in their operation. The direct-radiant sections are made of cast iron; the convector-type units may be of cast-iron sections having finned projections bounding air passageways at their rear, and those made of finned tubing may be placed within sheet-metal enclosures. The desirable placement of either of the two types is along the outside walls of the space to be heated. When because of inadequate wall space the required linear lengths of baseboard units cannot be placed along the bottoms of exterior walls the additional amounts are located along interior walls of the spaces to be heated.

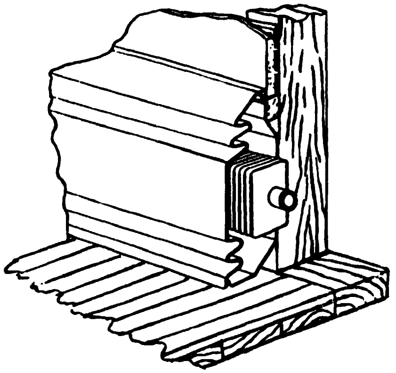
Figure 103 shows a hollow cast-iron unit which is $1\frac{3}{4}$ in. deep by 7 in. high and which is available in section lengths of 1 and 2 ft. The required amount of length, by one-foot increments, which replaces the room baseboard can be obtained by joining sections together with push nipples and bolt fastenings. This particular unit is suitable for use with any type of hot-water heating system as well as two-pipe steam and vapor plants. In order to take care of inequalities of floor level such equipment is hung on wall brackets so that vertical adjustment may permit the bottom edge to vary from nothing to $\frac{3}{8}$ in. above the floor. Air leakages behind the units may cause soiling and streaking of the walls above them; therefore a method of sealing is employed at both their tops and the floor level. This process consists in the attachment of tough paper on the surface of the wall behind them, $\frac{1}{2}$ in. of which is folded over the top edge of the metal. At the floor level and at the top of the unit further sealing against air leakage is



(Burnham Corp., Boiler Division)

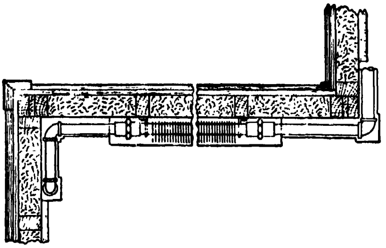
FIG. 104. Type-RC radiant baseboard.

secured by the use of quarter-round wood mouldings which are fastened in place by nailing so that the moulding butts against the already paper-sealed edges. An allowance of about $\frac{1}{8}$ in. in 10 ft for each 180 F temperature rise should be made for expansion of the metal. Fittings for corner connections of the sections and coverages of ends are available.



(*C. A. Dunham Co.*)

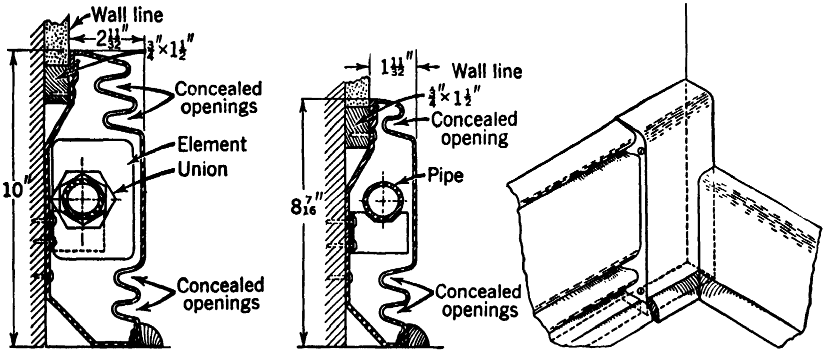
FIG. 105. Convector-type baseboard heating unit.



(*C. A. Dunham Co.*)

FIG. 106. Plan of pipe and finned sections of baseboard heating unit.

Finned radiant baseboard heaters are constructed in the form of the cast-iron unit illustrated by Fig. 104. These sections are 1 and 2 ft in length, 7 in. high, and 2 in. thick and are fastened together in the manner stated for the units illustrated by Fig. 103. A baseboard unit of the convector type which is built up of sheet metal and tubing is shown by typical views of its placement in Figs. 105 and 106.



Finned section Pipe section Corner section

(*C. A. Dunham Co.*)

FIG. 107. Details of convector-type baseboard heating unit.

Examination of Fig. 106 indicates that heat is obtainable from the plain-tube sections and from those heating elements which have finned sections. Details of the sectional dimensions of finned and pipe portions as well as corner details are given by Fig. 107. The finned units come in standard lengths of 2, 3, 4, 5, and 6 ft. The baseboard covering units for bare pipe sections have standard lengths of 10 ft and are cut to fit at the time of installation. The manufacturer supplies inside and outside corner fittings and those necessary at door openings; pipe and pipe fittings are those in ordinary commercial use. The details of installation as far as air sealing and circulation are given in Fig. 107. The air circulation is upward from the concealed opening at the bottom of the unit over the pipe and extended surfaces and out at the top.

170. Performances of Baseboard Heating Units. Tests have indicated that such equipment will operate to give a maximum temperature difference in the air between residence floor and ceiling levels of not to exceed 3 F when the outside-air temperature is 0 F. Manufacturer's data for the operation of plain direct radiant and convector-type cast-iron sections with water are given by the stated outputs for different conditions in Table 61.

TABLE 61

HEAT EMISSION DIRECT RADIANT AND FINNED CAST-IRON BASEBOARD SECTIONS*

Forced circulation of water Btu per hour per linear foot†

Water Temperature Deg F	Regular Radiant Baseboard	Finned Cast-Iron Baseboard
190	235	390
200	261	435
215	300	500

* Burnham Boiler Corporation. † Room air 70 F.

Tables 62 and 63 indicate the manufacturer's data for the performances of bare pipe and finned-heating elements made of tubing and sheet metal.

TABLE 62

HEAT OUTPUT OF ONE-INCH BARE PIPE IN DUNHAM BASEBOARD, BTU PER HOUR PER LINEAR FOOT

Water temperature, deg F	160	180	200	220
Heat output	76	96	117	140

Use of the data of Tables 62 and 63 is made in the following manner: the hourly heat losses from the room and the linear feet of

TABLE 63

HEAT OUTPUT OF FINNED SECTIONS OF DUNHAM BASEBOARD, BTU PER HOUR*

Cabinet Length, In.	Pipe Length, In.	Water Temperatures, Deg F 160	180	200	220
24	20	557	740	948	1171
36	32	937	1244	1594	1967
48	44	1317	1748	2239	2762
60	56	1697	2251	2884	3558
72	68	2077	2755	3528	4354

* C. A. Dunham Company.

exposed wall are ascertained and the operating temperature of the water fixed. The total heat output required is the sum of the heat emitted by the pipe installed and the additional output of the necessary finned sections. The manufacturer recommends a water temperature of 180 F as one which will not produce charring of air-borne dust particles.

PROBLEMS

1. A tubular radiator operated with saturated steam of 0.99 quality and 215 F temperature in a room where the air temperature was 70 F. The dry steam used per hour was 35 lb when the condensate left the radiator with the same temperature as that of the steam. Find the equivalent direct surface (edr) of the radiator in square feet and the weight of wet steam used per square foot of equivalent surface.

2. A building having heat losses amounting to 90,000 Btu per hr is to be maintained at 70 F by 4-tube 22-in. free-standing direct cast-iron radiators operating with saturated steam kept at a pressure corresponding to 215 F. How many radiator sections are necessary?

3. A 5-tube 25-in. steam radiator gave off 12,000 Btu per hr when operated under standard conditions of rating. Find the number of sections in the radiator.

4. How many 11B wall sections, Table 51, are required to supply 26,400 Btu per hr when operated with steam under standard conditions? Find the number of tubular wall sections, Table 52, are necessary to handle a load of 26,400 Btu per hr when operating under the standard conditions of steam rating. What output may be expected from the amounts of radiation just determined if in each case they are operated with gravity-flow hot water under standard conditions for water rating?

5. How many sections of small-tube wall radiation, Table 52, are required to emit 40,000 Btu per hr when operated as free-standing direct radiators in a room where the breathing-line air temperature is 70 F and the steam used for operation has a pressure of 15.65 psia?

6. Find the rated surface area and the dimensions of a free-standing cabinet steam radiator operated under its rating conditions to give 17,520 Btu per hr. What output will the radiator give when operated with gravity-flow hot water under standard conditions for water operation?

7. Calculate the heat given off by 50 sections of 3-tube 25-in. direct radiation when operated with steam at a temperature of 220 F and room air at 65 F.

8. A room having heat losses amounting to 21,000 Btu per hr is heated by a steam radiator operating with a pressure of 14 psia when the room temperature is 60 F. Find the number of 6-tube, 25-in. direct radiator sections used.

9. A hot-water heating plant is to operate with gravity flow of the water with the following temperature conditions; room air, 72 F; water entering radiator, 190 F; and water leaving radiator, 170 F. Find the heat output per square foot of rated surface of the radiator.

10. A convector, when operating with a steam temperature of 220 F and air entering the unit at 60 F, gave off 30,000 Btu per hr. Find the output of the unit under standard conditions of rating convectors.

11. A direct tubular steam radiator provides the capacity to overcome the losses of heat amounting to 22,000 Btu per hr from a room when it operates with a steam temperature of 218 F, a room temperature of 68 F, and an outdoor-air temperature of −20 F. What must be the steam temperature within the radiator if the room-air temperature is to be maintained at 68 F when the outside-air temperature is 20 F and the radiator is completely filled with steam? Assume that the room heat losses are directly proportional to the inside- outside-air temperature difference.

12. A room has heat losses of 8000 Btu per hr. Find the required linear length of a regular radiant baseboard and also a finned cast-iron baseboard when operated with water at a temperature of 200 F.

13. A baseboard heating unit is made up of 10 ft of one-inch pipe and 68 in. of finned pipe. What heat output may be expected per hour when operated with water at 180 F?

CHAPTER 8

HEATING BOILERS AND APPURTENANCES

171. Foreword. This chapter deals with the units used either in the production of low-pressure steam or in the heating of water for warming structures. Large-scale district heating installations, office buildings having power plants, and many industrial plants have steam generated at high pressures in power boilers of various types. The steam for heating purposes is reduced in pressure by passing it through pressure-reducing valves or steam engines or steam turbines. The engines and turbines abstract part of the heat energy of the steam for power purposes, and the low-pressure exhaust steam is suitable for radiators, convectors, blast units, and water heaters.

172. Distinctions between Steam Boilers and Hot-Water Heaters. The term hot-water boiler is applied to fuel-fired units for heating the water for hot-water heating systems. Such a water heater, completely filled with water, is usually identical in many respects with a steam heating boiler of the same type of construction. Any steam heating boiler may serve as a water heater if properly arranged and installed. Small round cast-iron boilers made for water heating do not as a rule have enough space in the top section for use as a steam boiler. The usual trimmings of a steam heating boiler include a pressure gage, a safety valve, a damper regulator, and a water column with a gage glass and try cocks. With some boilers a water column is not provided, and the gage-glass connections and the try cocks are attached directly to the boiler. The same unit when used as a water heater or a water boiler does not have a water column, gage glass, try cocks, or safety valve. The damper regulator is of a different type from that for a steam boiler. The gage attached to the heater is an altitude gage which may read either pounds per square inch pressure or the head of water in feet. A special type of thermometer is inserted in a tapping near the hot-water outlets to indicate the temperature of the heated water.

173. Materials of Construction and Working Pressures. Cast iron and steel are commonly employed in heating-boiler construction. Cast-iron boilers are built in sections which are assembled to produce a unit of the desired size. Steel boilers are constructed with a shell of steel plates either riveted or welded together and are provided with either fire tubes or water tubes which add to the water-heating surface. All cast-iron sectional boilers and many steel heating boilers have internal fireboxes or furnaces. As heating units in service they must operate for long periods of time between hand fuel firings and stokings and carry the load with the minimum amount of attention. Such operating conditions, therefore, presage low combustion rates on relatively large grate areas and a considerable fuel-storage capacity in the firepots when solid fuels are used.

The maximum safe working pressures usually specified for cast-iron boilers are 15 psig for steam and 30 psig for water-heating service. Low-pressure steel heating boilers usually have the same pressure limits as those stated for cast-iron construction. Steel boilers may be built for higher pressures than those stated previously, as closed systems of hot-water heating may involve pressures greater than 30 psig in the lowest part of the boiler or water heater.

174. Boiler Heating Surface. Those areas of a boiler which are in contact with the boiler water and which are exposed to either incandescent fuel, hot refractory materials, hot flue gases, or all the foregoing are termed water-heating surfaces. Some heating boilers are also arranged to have certain surfaces with steam on one side and hot gases on the other. Inasmuch as the heat transferred at such surfaces does not generally produce superheating of the steam, the areas involved are included with those of the water-heating surfaces in the estimation of the total heating surface of the unit. Some modern cast-iron heating boilers have sections with ribbed and extended areas which serve to increase their heating surfaces and keep the flue gases in contact with them.

The effectiveness of the heating surface is dependent upon the difference between the temperature of the fluid, water or steam, on one side, that of the gases on the other side, and the rate of circulation of both the water and the hot gases. These items affect the capacity which a boiler with a given amount of heating surface will develop.

Heating surfaces may be classified as direct and indirect. Direct surfaces are those upon which the light of the fire shines; they are very effective in the transfer of heat to the boiler water because of the high furnace temperatures which promote heat transfer both by radiation and convection. Indirect-heating surfaces are those in contact with flue gases only and are progressively less effective from the standpoint of heat transfer as the flue gases become cooled.

Additional heating surfaces are provided in boiler shells by the use of tubes. Fire-tube boilers are those in which the hot gases pass through tubes which are surrounded by water. The reverse is true in water-tube boilers where the water circulates through tubes which have their outer surfaces exposed to the travel of hot gases. Most boiler manufacturers use the outside diameters of the tubes employed in estimating the heating surfaces provided by the tubes.

The amount of heating surface provided in heating boilers per square foot of grate area is generally very much less than that provided in power-boiler installations. Power boilers usually have from 40 to 60 sq ft of heating surface per square foot of grate area. For heating boilers the ratio may be between 12 to 1 and 50 to 1, the greater number of boilers having heating-surface grate-area ratios varying between 18 to 1 and 30 to 1. The effect of a small ratio of heating surface to grate area is to lower the operating efficiencies when the boiler is driven hard to produce a given output.

175. Boiler Classifications. Heating boilers are built in a wide variety of designs. In the limited space here available it is impossible to describe all the different units available. One or more of the following classifications may be applied to any heating boiler:

Materials,	cast-iron and steel
Settings,	brick set and portable
Tubes,	fire tube and water tube
Flue-gas travel,	single pass and multiple pass
Draft,	direct up-draft and down-draft furnaces
Special,	boilers with a magazine feed for coal, etc. oil-burning units with an integral burner units especially designed for the burning of gas

Cast-iron boilers are never provided with brick settings, with the possible exception of a brick base, and generally the base is of cast iron. Certain steel boilers with internal fireboxes have a brick enclosure about them, and the gases having passed from the furnace through the fire tubes are made to flow over the outside of the shell within the brickwork. Steel boilers of the portable type have an internal firebox surrounded by water legs, and rest either on a cast-iron or a brick base beneath the firebox, the opposite end of the boiler being supported by a pedestal. These boilers have no enclosing brickwork.

In single-pass boilers the flue gases pass directly through the flues to the smoke outlet without reversal of direction of flow. When the flue gases from the furnace are made to travel back and forth through the gas passageways of the boiler the multiple-pass effect is secured. Such an arrangement may embody the principles of the horizontal return-tubular type of power boiler; it may involve passage of the gases from the furnace outlet at the rear of the boiler through flues to the front of the boiler and thence through other flues to the smoke outlet.

Direct updraft units have only a single grate in the firebox, and most of the air for combustion purposes passes upward from beneath the grate and through the fuel bed on the grate. The down-draft furnace has two grates, a water-cooled upper grate upon which the fuel is fired and a lower grate upon which fuel is not fired but which catches the drippings from the upper grate. A good fire of incandescent fuel is maintained on the lower grate, which receives its air supply through the ashpit beneath it. Air for combustion purposes is admitted above the green fuel on the upper grate and passes downward through it. The fuel placed on the upper grate burns on the lower side, and the volatile materials distilled from the fresh fuel and the smoke are carried downward over the hot fire on the lower grate where they are consumed. This arrangement promotes smokeless combustion.

Magazine boilers have a large fuel-carrying capacity and are so arranged that as the fuel is consumed it is replaced by fuel automatically moving down from the fuel-storage space of the boiler. Special oil-burning units have furnaces and heating surfaces adapted to the best utilization of oil as a fuel and are generally fitted with a burner and controls best suited to the boiler. Such a unit is usually very compact and is enclosed by an insulated casing. Special boilers for gaseous fuels are generally sectional, each section having its individual burner. Effort is always made to break up the flue gases into thin streams and to bring these streams into close contact with sufficient heat-absorbing surfaces to secure the maximum efficiency of operation. Ordinary heating boilers, both cast-iron sectional and steel units, may also be used with properly fitted oil and gas burners, and stokers.

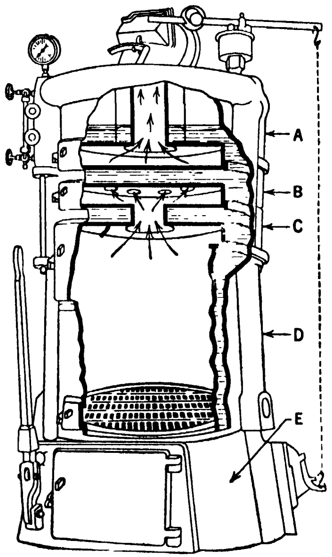


FIG. 108. Round sectional cast-iron steam boiler.

176. Cast-Iron Heating Boilers. These units are built in sizes ranging from very small up to those of moderate size capable of developing possibly 130 boiler horsepower. Cast-iron boilers are always constructed of sections joined together by some form of nipple or header connections. Cast-iron boilers may be further subdivided as those of the round type and those of the square type.

Round boilers vary somewhat in construction but in general are typified by the unit shown in Fig. 108. Such an assemblage has a top section *A* in which the outlets and safety-valve tappings are located, a number of intermediate sections *B*, *C*, depending upon the amount of heating surface required, a firepot or furnace section *D* in which the return tappings are located, and a base or ashpit section *E* which does not have any water space. The sections that confine the water or water and steam are fastened together with tapered nipples, and the sections under pressure are prevented from pulling apart by bolts or rods as indicated. Such units are small and compact and are built in sizes capable of supplying up to 1700 sq ft of standard steam radiation at their outlets. The return and outlet tappings are equal in number and size.

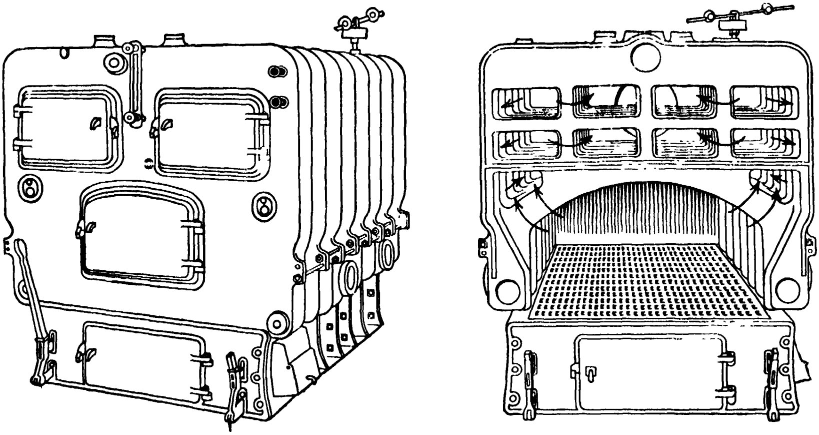
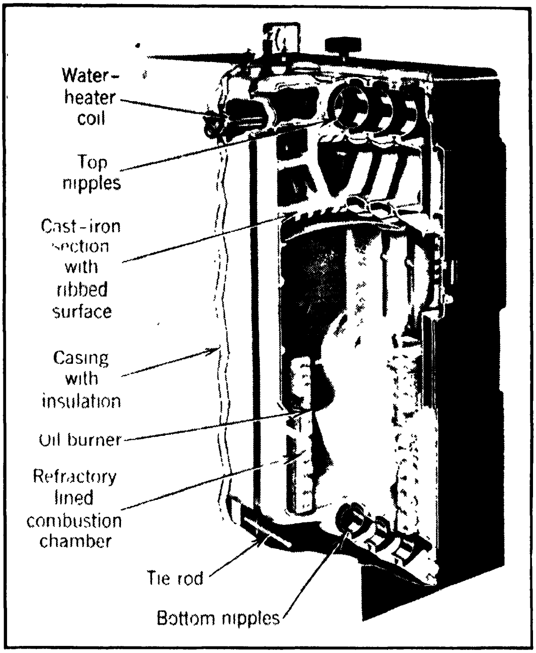


FIG. 109. Square sectional cast-iron steam boiler.

Square sectional boilers in general are of the order of the unit illustrated by Fig. 109. The assemblage consists of front and rear sections together with a number of intermediate sections depending upon the size of the boiler. Square sectional boilers range in size from very small units up to the largest sizes with 30 sq ft of grate area. The sections of large boilers may be composed of two pieces, and the various sections are always connected together at their tops and at their bottoms on each side, either by nipples of the push type or by an outside header at the top and at both sides of the bottom. When push nipples are used the sections are firmly held together by rods and nuts.

The number of top outlets to which piping is connected depends upon the size of the boiler and its method of fastening the sections together, i.e., either push-nipple or outside-header connections. The return tappings may be located in the rear section, in the sides of certain sections, both in the sides of the sections and the rear section, and in the return headers on both sides of the boiler when such headers are used.

Another example of a square-type cast-iron sectional boiler is the wet-bottom unit shown in Fig. 110. The boiler illustrated has a row of nipple connections between its sections both at their tops and their bottoms. Another form of wet-bottom boiler construction has two rows of connecting nipples at the bottoms of the boiler sections and one row at their tops. The cast-iron boiler of Fig. 108 has a separate base which does not confine water and requires a floor of fireproof construction beneath it. When a wet-bottom boiler is used with either oil or gas as a fuel the furnace is completely surrounded by water-filled portions of its sections; those units which operate burning hand-fired solid fuels have water-cooled ashpit bottoms and sides. Wet-bottom boilers are built in relatively small units for both steam and water service and may be installed upon wooden floors. One



(*National Radiator Co.*)

FIG. 110. Wet-base sectional cast-iron hot-water boiler.

field of application of wet-bottom boilers is in hot-water heating systems placed in residences which do not have basements.

All heating boilers should be well insulated. This insulation may consist of either plastic material applied to the boiler or a blanket of insulation, such as mineral wool, which is covered with an enameled steel jacket. Plastic insulation may be either troweled smooth and left without further finish or it may be covered with canvas and painted.

177. Capacity Ratings of Cast-Iron Boilers. The manufacturers of such equipment express the output of the units in terms of either Btu per hour or square feet of standard radiation. This latter item may be square feet of standard steam radiation, 240 Btu per hr, and square feet of standard hot-water radiation, 150 Btu per hr per sq ft. Care must always be taken to note whether the output given is that at the boiler outlets or nozzles or whether the capacity given is that which the boiler is to supply in the building when certain allowances have been made for piping losses and other factors.

Many manufacturers list a part or all of their boilers with data pertinent to I = B = R gross output and I = B = R net rating in square feet of steam and hot-water radiation, either of which can be expressed in terms of Btu per hour. The code[1] of the Institute of Boiler and Radiator Manufacturers designates prescribed methods for testing and rating cast-iron heating boilers, except those of the magazine type, for conditions of both hand and automatic firing.

Hand-fired boilers under test must develop an overall efficiency of 58 per cent or more with anthracite coal having a heating value of 13,000 Btu per lb on a dry basis. Limits are set for chimney area and height, draft in the stack, and the minimum time that an available fuel charge will last when it is burned with a combustion rate sufficient to produce the listed I = B = R gross output. When the fuels are either bituminous coal or coke the same specifications as regards heating value of the fuel and the other limitations apply.

Oil-burning boilers must operate with a minimum overall efficiency of 68 per cent and a maximum temperature of 600 F for the flue gases leaving the boiler; the burner adjusted to give 10 per cent CO_2 (with variations of ±0.2 per cent); heat release in the furnace not to exceed 80,000 Btu per hr per cu ft; and draft losses not to be greater than a designated amount. Stoker- and gas-fired units are rated with the same output as oil-fired boilers unless the gas-fired boilers have been

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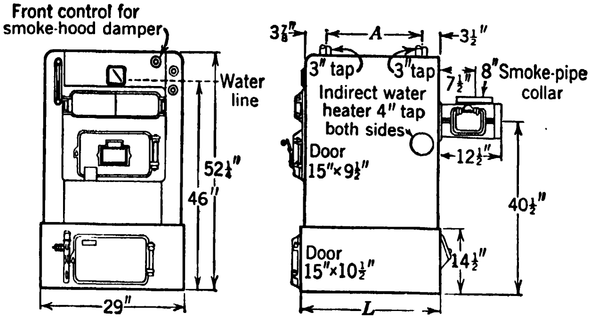
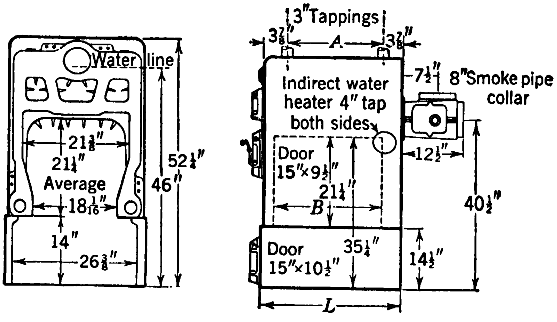
[1] "I = B = R Testing and Rating Code for Low Pressure Heating Boilers," Third edition, the Institute of Boiler and Radiator Manufacturers, 60 East 42nd Street, New York, N. Y., July 1945.



Hand firing



Automatic firing

Boiler Number, Steam or Water	Number and Size of Outlets, In.	Number and Size of Inlets, In.	Dimension, In.			Base Inside Length,† In.	Number of Grate Bars	Section Assembly A—Front with 1-3-in. flow tap B—Back with 1-3-in. flow and 2-3-in. return taps CT—Intermediate with safety valve and regulator tap C—Intermediate, plain
			A	B	L			
167*	2-3	2-3	14	15½	21¼	19¾	4	167*—A-CT-B
267*	2-3	2-3	18	19½	25¼	23¾	5	267*—A-C-CT-B
367*	2-3	2-3	22	23½	29¼	27¾	6	367*—A-C-C-CT-B
467*	2-3	2-3	26	27½	33¼	31¾	7	467*—A-C-C-C-CT-B

* Substitute *S* for steam, *W* for water.

† Base inside, width between ribs 26⅜ in., height 14 in. Length of bars for hearth support 28¼ in.

(*Weil-McLain Co.*)

Fig. 111. Dimensions of sectional cast-iron boilers.

rated by the American Gas Association. Under such conditions the AGA rating is used.

The I = B = R gross-output rating is the total output of the boiler. while the I = B = R net rating is the capacity that the boiler may supply when allowances have been made for piping losses of the system and a pickup factor. For hand-fired units the piping losses and the pickup factor together range from 2.36 for boilers having a gross I = B = R rating of 56,600 Btu per hr to 1.40 for boilers having a gross I = B = R rating of 6,720,000 Btu per hr. For automatically fired boilers the combined factors range from 1.56 to 1.288 when the net capacities are from 100 to 20,000 sq ft of steam edr. Thus a hand-fired boiler having a gross I = B = R output of 6,720,000 Btu per hr has a net I = B = R rating of 6,720,000 ÷ 1.40 = 4,800,000 Btu per hr or 20,000 sq ft of steam edr. For an automatically fired installation the gross I = B = R output to give the net rating of 20,000 sq ft of steam edr is 6,182,000 Btu per hr. Numerically 6,182,000 ÷ 1.288 = 4,800,000 Btu per hr or 20,000 sq ft of standard steam radiation.

For boilers not rated by the foregoing I = B = R methods the Heating, Piping and Air Conditioning National Association expresses ratings based on their physical characteristics. These net steam ratings are made on the basis that each square foot of grate area will always give a definite amount of capacity. The American Gas Association ratings are based on performance tests. The ASHVE has a number of codes which outline the methods of testing hand-, stoker-, and oil-fired boilers. From the data of tests run under the prescribed conditions the operating efficiencies and performances of heating boilers may be ascertained.

178. Ratings and Physical Dimensions of Commercial Cast-Iron Boilers. The ratings of Table 64 and the dimensions and other items indicated by Fig. 111 are typical of the data in catalogs supplied by the builders of cast-iron heating boilers which have been given I = B = R ratings.

179. Advantages and Disadvantages of Cast-Iron Boilers. The construction of sectional boilers allows the various parts to be taken through ordinary doorways for assembly in the boiler room. This is a distinct advantage both in a new installation and in the repair of damaged boilers. Cast iron resists the action of corrosive agents much better than steel. Many cast-iron steam boilers have a low water line, which is a great advantage, especially where gravity-return steam-heating systems are used, as structural conditions of the boiler room may preclude the use of a boiler with a high water level or else make a boiler pit necessary.

TABLE 64

Capacity Ratings of Well-McLain Number 67 Series Cast-Iron Square Sectional Boilers for All Fuels

Hand Fired

Boiler Number Steam or Water	Net I = B = R Ratings* †			Gross I = B = R Output, Btu per Hr	Grate Area, Sq Ft	Chimney	
	Steam, Sq Ft	Water, Sq Ft	Btu per Hr			Size, In.	Height, Ft
167-‡	290	465	70,000	162,000	2.38	8 × 8	30
267-‡	380	610	91,000	208,000	2.91	8 × 8	35
367-‡	470	755	113,000	252,000	3.44	8 × 12	35
467-‡	560	900	134,000	295,000	3.97	8 × 12	40

Stoker Fired

Boiler Number Steam or Water	Net I = B = R Ratings* †			Gross I = B = R Output Btu Per Hr	I = B = R Burner Capacity, Coal, Lb Per Hr	Chimney	
	Steam, Sq Ft	Water, Sq Ft	Btu per Hr			Size, In.	Height, Ft
K-267-‡	440	705	106,000	161,000	18.0	8 × 8	35
K-367-‡	530	850	127,000	192,000	21.0	8 × 12	35
K-467-‡	620	990	149,000	223,000	25.0	8 × 12	35

Oil Fired

Boiler Number Steam or Water	Net I = B = R Ratings*			Gross I = B = R Output, Btu Per Hr	I = B = R Burner Capacity, Oil Gal Per Hr	Chimney	
	Steam, Sq Ft	Water, Sq Ft	Btu per Hr			Size In.	Height, Ft
0-167-‡	350	560	84,000	129,000	1.25	8 × 8	30
0-267-‡	440	705	106,000	161,000	1.60	8 × 8	35
0-367-‡	530	850	127,000	192,000	1.95	8 × 12	35
0-467-‡	620	990	149,000	223,000	2.25	8 × 12	35

* Net I = B = R ratings are based on net installed radiation of sufficient quantity for the requirements of the building, and nothing need be added for normal piping and pickup. No allowance has been made for domestic water heating.

† For coal of 13,000 Btu per lb.

‡ Substitute *S* (for steam), *W* (for water).

The disadvantages of cast-iron boilers include danger of being damaged when improperly handled or when used with poorly installed steam-heating systems, small steam and water-storage spaces, and the likelihood of discharging wet steam and slugs of water when driven hard.

180. Steel Heating Boilers. Heating boilers constructed of steel involve several arrangements. The categories which may be included are: brick set, portable, fire-tube, water-tube, down-draft, up-draft, single-pass and multiple-passes for the flue gases, and vertical and horizontal.

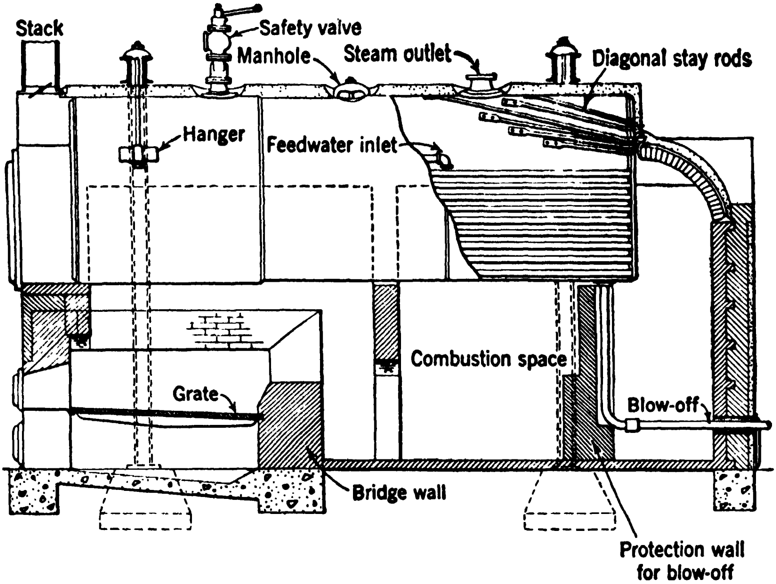


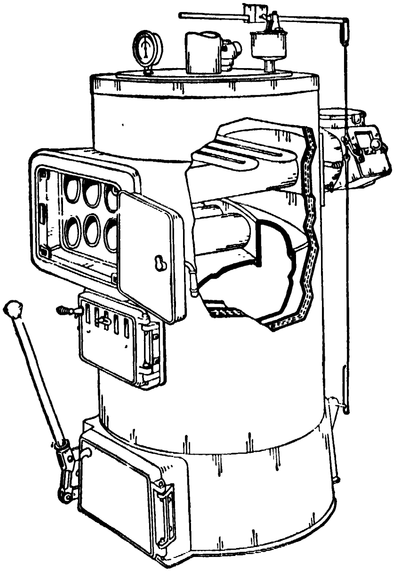
FIG. 112. Horizontal return-tubular boiler.

181. Horizontal Return-Tubular Boilers. A longitudinal view of a return-tubular boiler and setting is shown by Fig. 112. This type of boiler, which may be used for either power purposes or steam heating, is brick set and is a fire-tube unit. The furnace, which is external to the shell, consists of the grates, the bridge wall, and the combustion space. The ashpit is beneath the grate. Access to the furnace and the ashpit is through the firing and ashpit doors located in the boiler setting front. The front and sides of the furnace are lined with firebrick. The boiler shell is most satisfactorily supported by steel columns and hangers as shown by Fig. 112. Small units may be supported directly by the brickwork of the setting.

The hot gases pass over the bridge wall along the under side of the shell to the rear of the setting and then through the fire tubes in the lower part of the shell to the front of the boiler. From the front end of the tubes the gases pass to the smokebox and then into the breeching connected to the stack.

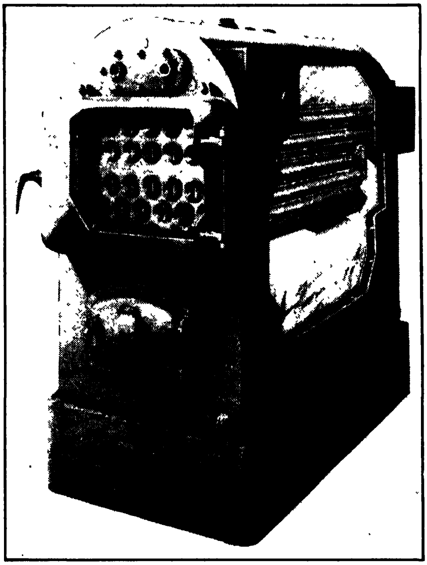
These units are built in sizes ranging from 36 in. in diameter and 8 ft long to 84 in. in diameter and 20 ft long. The maximum working pressure for power purposes ranges up to 175 psig and that for heating purposes alone up to 15 psig. The horsepower capacities vary from 15 to 350, and the capacities in square feet of standard steam radiation range from 2100 to 45,000 at the boiler outlet.

182. Small Round and Rectangular Steel Boilers. The small steel boilers of Figures 113 and 114 embody in their construction the welding together of steel plates and tubes. The round boiler of Fig. 113 is suitable for small residences and comparable structures. The rectangular boiler of Fig. 114 is adapted to usage in large residences, stores, garages, and churches. The round boilers are usually fitted with an insulated jacket; the square boilers may or may not have a surrounding jacket. These



(Kewanee Boiler Corp.)

Fig. 113. Small round steel boiler.



(Kewanee Boiler Corp.)

Fig. 114. Square type-R steel heating boiler.

TABLE 65

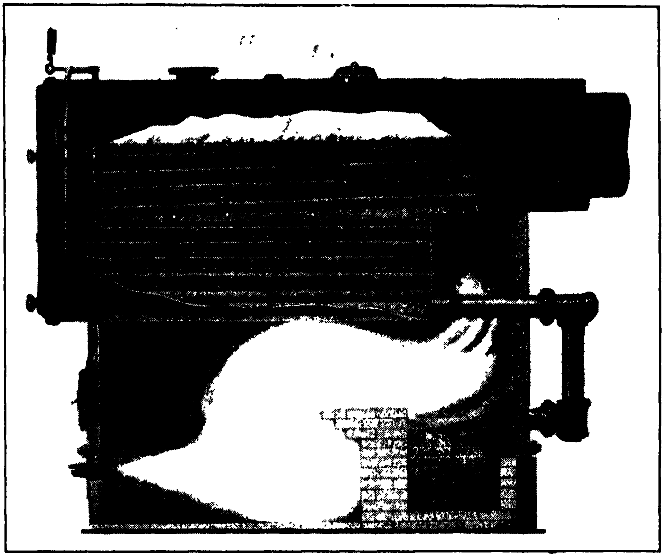
RATINGS AND SPECIFICATIONS OF KEWANEE "SQUARE HEAT" 3R STEEL BOILERS

Specifications		Oil Gas or Stoker								Hand-Fired Coal							
Boiler number		3R1	3R2	3R3	3R4	3R5	3R6	3R7	3R8	3R1	3R2	3R3	3R4	3R5	3R6	3R7	3R8
SBI net rating, steam	sq ft	900	1100	1300	1500	1800	2200	2600	3000	740	910	1080	1230	1480	1810	2140	2480
water	sq ft	1440	1760	2080	2400	2880	3520	4160	4800	1180	1450	1720	1970	2370	2890	3430	3970
Btu per hour	1000's	216	264	312	360	432	528	624	720	178	218	261	295	355	434	514	595
Heating surface (SBI minimum)	sq ft	53	65	77	88	106	129	153	177	53	65	77	88	106	129	153	177
Furnace volume (SBI minimum)	cu ft	8.2	10.0	11.8	13.6	16.4	20.0	23.6	27.3								
Grate area (SBI minimum)	sq ft									4.2	4.9	5.7	6.3	7.2	8.4	9.4	10.5
Boiler-shell width overall	in.	30	30	30	30	34	34	34	34	30	30	30	30	34	34	34	34
Boiler-shell length	in.	30½	36½	42½	48½	42½	50½	58	67	30½	36½	42½	48½	42½	50½	58	67
Boiler height overall, from floor	in.	65	65	65	65	72½	72½	72½	72½	65	65	65	65	72½	72½	72½	72½
Water-line height	in.	55½	55½	55½	55½	62½	62½	62½	62½	55½	55½	55½	55½	62½	62½	62½	62½
Shipping weight unjacketed	lb	1500	1650	1800	1950	2250	2475	2725	3000	1775	1975	2175	2350	2600	2900	3200	3500
Outside surface to cover	sq ft	38	43	48	53	58	65	73	82	38	43	48	53	58	65	73	82

Steam-supply size 4 in. Base height 14 in.

boilers are updraft units in which the flue gases make two passes through the firetubes. Rating and other data are given for small rectangular units in Table 65.

183. Portable Steel Boilers. The oil-fired steel boiler of Fig. 115 has an internal firebox or furnace surrounded by water legs. The boiler illustrated is an updraft unit with two passes for the flue gases through its firetubes. These boilers rest on either a steel or a brick base and have no outside brickwork. Refractory brickwork is installed within the base in the oil-burning unit shown by Fig. 115. Rated capacities and physical dimensions of the boiler of Fig. 115 are given in Table 66 for both hand-fired and automatically fired operations.



(Pacific Steel Boiler Division, United States Radiator Corp.)

FIG. 115. Oil-fired steel heating boiler.

184. Ratings of Steel Boilers. The Steel Boiler Institute, Inc., a manufacturers' organization, has a code which embodies specific directions for the testing of and the empirically rating of both hand-fired and mechanically fired steel heating boilers. The code classifies steel boilers into two general groups which are *commercial* and *residential.* The term commercial applies to large-size boilers while the designation residential size is given to those units which have less than 177 sq ft of heating surfaces and which have steam ratings of not more than 2480 sq ft edr when hand fired and 3000 sq ft when mechanically fired. As with other boilers the emission of 240 Btu per hr with steam and the output of 150 Btu per hr with hot water constitutes one square foot of equivalent direct radiation, edr. The rating of a boiler in terms of hot-water capacity is $\frac{240}{150}$ times its steam capacity. For either commercial or residential hand-fired steel boilers the net steam ratings are not to exceed 14 times their heating surfaces and when stoker, oil, or gas fired the steam ratings shall not exceed 17 times their heating surfaces, which are always expressed in terms of square feet. The ratings for hand-fired units are made on the basis of coal having a heating value of 13,000 Btu per lb as fired. When the fuel used has a heating value less than 13,000 Btu per lb the rating of a given boiler must be reduced accordingly or else a boiler with a greater rated capacity must be chosen to care for a given load.

In connection with boiler-capacity listings two terms are used; these are SBI Rating and SBI Net Rating. The SBI Rating denotes the

TABLE 66

MEASUREMENTS AND RATINGS OF PACIFIC STEEL HEATING BOILERS FOR HAND OR AUTOMATIC FIRING

Hand Fired

Catalog Number	SBI Rating, Sq Ft Steam	Height Water Line, In.	One Boiler		Two Boilers		Grate Area, Sq Ft	Heating Surface, Sq Ft	Size Outlet, In.	Size Return, In.
			Diam. Stack, In.	Height Stack, Ft	Diam Stack, In.	Height Stack, Ft				
2922	2,200	59½	15	45	20	55	8.9	158	6	3
2923	2,600	59½	16	50	22	60	9.7	186	6	3
2924	3,000	59½	16	50	22	60	10.5	215	6	3
3321	3,500	68½	17	55	23	65	11.4	250	6	3
3322	4,000	68½	18	55	24	65	12.2	286	6	3
3323	4,500	68½	19	55	26	65	13.4	322	6	3
3324	5,000	68½	20	60	27	70	14.5	358	6	3
4021	6,000	68	21	60	28	70	16.4	429	8	4
4721	7,000	78	22	65	29	75	18.1	500	8	4
4722	8,500	78	24	65	32	75	20.5	608	8	4
5221	10,000	82	26	70	35	80	22.5	715	8	4
5222	12,500	82	28	75	37	85	25.6	893	8	4
6021	15,000	90	29	75	39	85	28.4	1072	8	4
6022	17,500	90	31	80	41	90	30.9	1250	8	4
6721	20,000	98½	33	80	44	90	33.2	1429	8	4
6722	25,000	98½	36	95	48	105	37.4	1786	10	6
7521	30,000	106	38	95	51	105	41.2	2143	10	6
7522	35,000	106	40	120	53	130	44.7	2500	10	6

TABLE 66 (*Continued*)
Automatically Fired

Catalog Number	SBI Rating, Sq Ft Steam	Height Water Line, In.	One Boiler		Two Boilers		Gross Base Volume, Cu Ft	Firebox Volume		Heating Surface, Sq Ft	Size Outlet, In.	Size Return, In.
			Diam. Stack, In.	Height Stack, Ft	Diam. Stack, In.	Height Stack, Ft		Above Base, Cu Ft	SBI Minimum Cu Ft			
2982	2,680	59½	15	35	20	45	11	24	19.2	158	6	3
2983	3,160	59½	16	35	22	45	13	28	22.6	186	6	3
2984	3,650	59½	16	40	22	50	15	32	26.1	215	6	3
3381	4,250	68½	17	45	23	55	15	34	30.4	250	6	3
3382	4,860	68½	18	45	24	55	17	39	34.8	286	6	3
3383	5,470	68½	19	45	26	55	19	43	39.1	322	6	3
3384	6,080	68½	20	50	27	60	21	47	43.5	358	6	3
4081	7,290	68	21	50	28	60	27	69	52.1	429	8	4
4781	8,500	78	22	50	29	60	26	78	60.8	500	8	4
4782	10,330	78	24	60	32	70	31	93	73.8	608	8	4
5281	12,150	82	26	55	35	65	34	106	86.8	715	8	4
5282	15,180	82	28	65	37	75	42	129	108.5	893	8	4
6081	18,220	90	29	65	39	75	43	142	130.2	1072	8	4
6082	21,250	90	31	70	41	80	51	168	151.8	1250	8	4
6781	24,290	98½	33	70	44	80	67	192	173.5	1429	8	4
6782	30,360	98½	36	85	48	95	83	236	216.9	1786	10	6
7581	36,430	106	38	80	51	90	87	255	260.3	2143	10	6
7582	42,500	106	40	100	53	110	101	295	303.6	2500	10	6

estimated design load that a boiler will carry. This load is the sum of the following items: (*a*) the calculated square feet of equivalent direct radiation necessary to heat the building as determined by commonly accepted methods of estimation; (*b*) the maximum water heating load and that of other apparatus connected to the boiler, square feet edr; and (*c*) the estimated heat emission of piping to radiators and other apparatus connected to the boiler expressed in terms of equivalent direct radiation. Each of the foregoing items as well as their sum may be expressed in heat units per hour. The SBI Net Rating is the net load the boiler will carry as represented by the sum of (*a*) and (*b*), except that when the heat losses from the piping exceed 20 per cent of the installed radiation the excess is considered as additional net load.

In general the volume of furnaces in which solid fuel is burned is the cubical content of the space between the bottom of the fuel bed and the first plane of entry into or between the tubes. For those furnaces in which either pulverized, liquid, or gaseous fuels are burned the volume is that between the hearth and the first plane of entry into or between the tubes. Grate area, expressed in square feet, is measured in the plane of the top surface of the grate. In double-grate boilers the total grate surface is taken as the sum of that of the upper grate and one-fourth of that of the lower grate. The following equations apply to commercial boilers in the determination of the proper grate area G, in square feet, based on SBI ratings of R sq ft edr. For boilers having R ranging from 1800 to 4000 sq ft edr steam,

$$G = \sqrt{\frac{R - 200}{25.5}} \tag{88}$$

and for those with 4000 sq ft of steam edr or greater

$$G = \sqrt{\frac{R - 1500}{16.8}} \tag{89}$$

Stoker-fired bituminous coal, pulverized-fuel, oil-, and gas-burning commercial boilers must have furnace volumes which have not less than one cubic foot of space for each 140 sq ft of SBI steam rating. No limitation is placed on the volume of the furnace of a mechanically fired anthracite-coal-burning boiler. The furnace height of a commercial boiler is the vertical distance from the bottom of the water leg to the crown sheet, measured between the sidewalls and midway between the front and the back (bridge wall when used) of the furnace. The code provides data for minimum heights for various capacities and when necessary the boiler is to be set upon an extended base to secure the required height.

Residential oil-fired boilers shall operate with a burner set to produce 10 per cent CO_2 (± 0.20 per cent) in the flue gases, and when operating at 150 per cent of the SBI net rated capacity the temperature of the escaping flue gases shall not exceed 600 F, the combined efficiency of the burner and the boiler shall not be less than 70 per cent, and the draft differential between the breeching and the firebox shall not be more than $DL = (NR/200) + 4$ where DL is the draft loss in hundredths of an inch of water and NR is the SBI net rating in square feet of steam radiation. This limitation does not apply to boiler-burner units integrally constructed with refractories and provided with means to overcome higher draft losses. A minimum furnace volume of one cubic foot per 110 sq ft of steam SBI net rating is also necessary except in those boiler-burner units of integral construction. Stoker- and gas-fired units shall not have ratings in excess of those for oil-fired boilers.

The following equation gives an expression for the grate area G, in square feet, for a hand-fired residential steel heating boiler

$$G = \sqrt{\frac{NR + 600}{9}} - 8 \tag{90}$$

where NR is the SBI net rating in square feet of steam radiation. The firebox volume of a residential boiler used in determining coal-storage capacity is the volume between the grate and the crown sheet. The volume of the uptake, the upper combustion chamber, and that beyond the front face of the bridge wall is not included. The firebox volume FV, expressed in cubic feet, is not to be less than

$$FV = \sqrt{\frac{NR + 90}{5}} - 5.5 \tag{91}$$

185. Heating Boiler Tests. The operating characteristics and the performances of heating boilers can best be determined as the result of actual tests in which sufficient and accurate observations and data are taken. The details of the necessary equipment, the methods of procedure, the observations to be made and all the calculations involved in the testing and determination of the performance of a heating boiler are beyond the scope of this book. The reader is referred to the various existent codes[2, 3, 4, 5, 6] for testing and rating heating boilers. Boiler performances, as determined by tests under various operating conditions, can be shown graphically by curves similar to those of Fig. 116.

186. Boiler Capacity and Efficiency. The capacity of a steam boiler equals the hourly heat input to the delivered steam formed divided by 240. This expression gives the capacity in square feet of standard steam radiation. For hot-water heaters the capacity in square feet of standard water radiation is obtained by dividing the hourly heat output by 150.

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[2] "I = B = R Testing and Rating Codes for Low Pressure Heating Boilers," 1945.

[3] "ASHVE Standard and Short Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers, Codes 1 and 2, 1929.

[4] "ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers," Code 3, 1929.

[5] "ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel," 1932.

[6] "ASHVE Standard Code for Testing Stoker-Fired Steam-Heating Boilers," 1938.

The hourly heat output of a steam boiler is

$$H = (h_f + xh_{fg} - h_{f1})W_w \tag{92}$$

where h_f = enthalpy of the liquid of the steam at the absolute boiler pressure, Btu per lb.
x = quality of the steam at the boiler outlets.
h_{fg} = enthalpy of evaporation of the steam at boiler pressure, Btu per lb.
h_{f1} = enthalpy of the liquid of the feedwater, Btu per lb.
W_w = weight of water evaporated per hour, lb.

The capacity of a steam boiler in terms of square feet of standard steam radiation is

$$S = \frac{(h_f + xh_{fg} - h_{f1})W_w}{240} \tag{93}$$

Manufacturer's catalogs usually state capacities for boilers in terms of square feet of both steam and hot-water equivalent direct radiation. The water capacities so listed are very close to $\frac{240}{150}$ times the steam capacities. The automatically-fired boiler $3R1$ of Table 65 has a rated steam capacity of 900 sq ft and a water capacity of 1440 sq ft edr, which is equal to $900 \times \frac{240}{150}$.

For boilers tested as hot-water heaters the total heat output H per hour is

$$H = (h_{f2} - h_{f1})W_w \tag{94}$$

where h_{f1} = enthalpy of the liquid of the water entering the boiler, Btu per lb.
h_{f2} = enthalpy of the liquid of the water leaving the boiler, Btu per lb.
W_w = weight of water heated per hour, lb.

For a water boiler the capacity in square feet of radiation is

$$S = \frac{(h_{f2} - h_{f1})W_w}{150} \tag{95}$$

The overall efficiency E_b of a boiler unit, including the furnace and grates, when grates are required, is the ratio of the heat output to the heat of the fuel charged to the furnace in a given interval of time. Thus for a steam boiler the overall efficiency E_b in per cent is

$$E_b = \frac{(h_f + xh_{fg} - h_{f1})W_w}{W_f \times F} \times 100 \tag{96}$$

and for a hot-water heater or boiler

$$E_b = \frac{(h_{f2} - h_{f1})W_w}{W_f \times F} \times 100 \tag{97}$$

where W_f = weight of fuel fired per hour, lb.

F = heating value of fuel as fired, Btu per lb.

When either gas or oil is used as a fuel the total quantity in the denominators of equations 96 and 97 is the units of fuel used per hour times the heating value of the fuel per unit.

The overall efficiency generally increases as the capacity developed increases until a maximum efficiency is obtained at some load and then decreases as the load is further increased. The decrease of efficiency after the maximum is reached is due to the inability of the heating surfaces to absorb the heat with resultant higher outlet flue-gas temperatures and greater heat losses to the stack. This has an important bearing in the selection of a heating boiler, for if a boiler is selected on the basis of a high rated capacity it may be uneconomical because of its low operating efficiency. Heating boiler overall efficiencies range from 50 to 70 per cent.

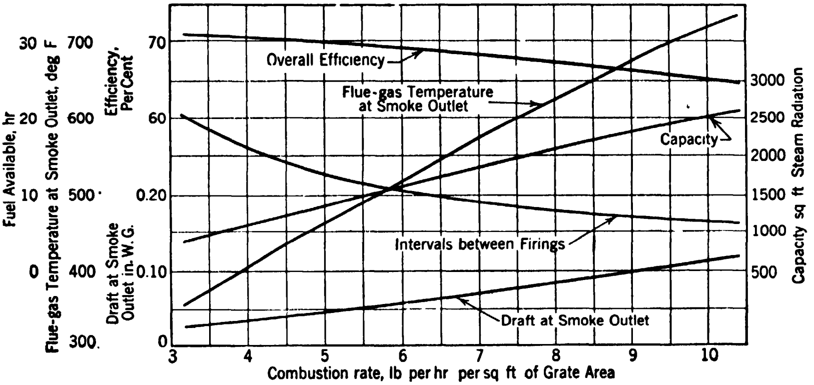


FIG. 116. Performance curves for a hand-fired cast-iron sectional heating boiler. Grate area 7.45 sq ft, heating value of coal 12,500 Btu per lb.

Figure 116 gives the performance curves for a typical hand-fired solid-fuel-burning heating boiler when operated with various outputs. Note that as the output is increased the time that the fuel charge will last is decreased, the draft requirements become greater, and the temperatures of the escaping flue gases rise.

187. Boiler Horsepower. Sometimes it may be desirable to express the capacity of a steam boiler in terms of boiler horsepower as well as in terms of standard square feet of radiation. In the calculation of boiler horsepower a term known as equivalent evaporation may be used. Equivalent evaporation is the pounds of dry steam that would have been formed by the heat added had the feedwater been supplied at a temperature of 212 F and converted into steam at the same temperature. Thus the evaporation of the water would have been from and at 212 F, and the heat added per pound of water would have been the latent heat of evaporation at 212 F. The latter quantity is 970.3 Btu per lb. The equivalent evaporation per pound of feedwater evaporated under actual conditions is the factor of evaporation *FE* and is

$$FE = \frac{h_f + xh_{fg} - h_{f1}}{970.3} \tag{98}$$

The heat symbols of equation 98 are the same as those of equation 92.

A boiler horsepower is defined as the equivalent evaporation of 34.5 lb of water per hr from and at 212 F, and its heat equivalent is 34.5 × 970.3 = 33,475 Btu per hr.

$$\text{Boiler horsepower} = \frac{(h_f + xh_{fg} - h_{f1})W_w}{34.5 \times 970.3} \tag{99}$$

The equivalent of a boiler horsepower in terms of standard steam radiation is 33,475 ÷ 240 = 139.5 sq ft at the boiler nozzles or outlets.

188. Boiler Heating Load. The proper selection of a boiler or boilers for a heating installation cannot be made until the total load imposed at the boiler outlets is determined. This load may be expressed in either Btu per hour or the equivalent square feet of direct radiation surface that must be supplied.

The maximum boiler load includes:

1. The heat supplied to all heated spaces by all forms of heat-emitting units such as direct radiators, convectors, unit heaters, unit ventilators, and hot-blast systems. This load is for the plant design conditions with respect to both indoor- and outdoor-air temperatures.
2. Heat lost from all steam and water piping both insulated and uninsulated.
3. The loads imposed by water heaters, steam tables, etc. This class of service may be a very heavy one, and strict attention must invariably be given to its magnitude.
4. The heat necessary to bring the building walls and contents, including the radiators and piping, up to the working temperatures when the structure is allowed to cool during the night or between periods of usage when the heating is intermittent.

5. The extra heat given off by the piping and heat disseminators during warming-up periods due to the reduced room temperatures.

The total of the first three items is the normal design load that the boiler plant must handle in the coldest weather when the heating operations are continuous, 24 hr per day. If the building is operated with reduced air temperatures during the night or if the building is heated intermittently, i.e., only one, two, or three days per week, as may be true of a church, then items 4 and 5 must be added as a starting-up allowance to the normal design load. Warming-up allowances to be added to the design load as recommended by the American Society of Heating and Ventilating Engineers are given by Table 67.

TABLE 67

WARMING-UP ALLOWANCES FOR LOW-PRESSURE STEAM AND HOT-WATER HEATING BOILERS* †

Design Load Representing Summation of Items 1, 2, and 3‡		
Btu per Hr	Equivalent Sq Ft of Steam Radiation‡	Percentage Capacity to Add for Warming Up
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2,500	55
600,000 to 1,200,000	2,500 to 5,000	50
1,200,000 to 1,800,000	5,000 to 7,500	45
Above 1,800,000	Above 7,500	40

* Copyright, American Society of Heating and Ventilating Engineers, ASHVE Code of Minimum Requirements for the Heating and Ventilation of Buildings, 1929.

† This table refers to hand-fired solid-fuel boilers. A factor of 25 per cent over design load is adequate when oil or gas are used as fuels.

‡ 240 Btu per sq ft.

The estimated design load in terms of equivalent direct steam radiation is the load in Btu divided by 240. Whenever possible the water-heating load should be based on the pounds of water heated per hour and its temperature rise. If the water-heating load cannot be so calculated the following approximations may be used with discretion. Thus for water heaters, having firebox coils, the tank storage capacity in gallons is multiplied by 2 to approximate the square feet of equivalent direct steam radiation. When hot water from the boiler is passed through the heater $\frac{1}{2}$ sq ft of steam radiation may be allowed for each gallon of water heated per hour. The equivalent of direct-indirect radiator surface is taken as 1.25 times the rated surface.

Data relative to the calculation of piping losses are available in Chap. 9. Sometimes a flat allowance of 15 to 40 per cent in addition to the installed load is made to cover piping and boiler losses. The smaller allowance is made for very large systems, the larger allowance for very small plants, and intermediate allowances for systems between very large and very small in size.

189. Factors Affecting Boiler Selection. Local operating conditions largely influence the selection of a boiler. These include the nature of the service to be supplied, the chimney draft, the fuel, the boiler attendance, reserve capacity necessary, the possibility of the need for future extensions, and the amount and character of heating load, i.e., continuous or intermittent. Because of the necessity of the use of certain fuels some boilers may not be suitable, and others may not operate successfully with the draft available. For some boilers, such as those for hospital work, 100 per cent reserve capacity may be necessary. Consideration should be given to the desirability of providing two or more boilers in order to produce flexibility of operation and some reserve capacity in each boiler which may be available should a unit fail.

190. Selection of Solid-Fuel-Burning Boilers. Article 188 indicates that the total load imposed on any boiler may accrue from two or more demands for heat. Therefore, in considerations of the ratings and capacities of both cast-iron and steel boilers and their applications in the selection of suitable and proper units, care must be taken to distinguish between such terms as gross output, net output, net load, design load, allowances made for piping losses, water-heating loads, and pick-up factor. Irrespective of how a boiler may be rated it must when in service be able to supply at its outlet nozzles the amount of heat necessary to satisfy all requirements made of it by the attached piping and equipment. When the factors involved are known the selection of a boiler may be made on one of the following bases:

a. Published ratings given by the builder.

b. Capacities based on physical characteristics of the boiler.

c. Information obtained from actual output tests of the type of boiler desired.

d. The required grate area of a hand-fired unit when due consideration is given to the heating surfaces, probable combustion rate, and expected operating efficiency.

Method a may be followed if the exact basis of rating is known and if the published ratings are conservative so that low operating efficiency will not be concurrent with operation at the rated capacity. Uncertainty often exists in regard to the various allowances which may or may not have to be made relative to piping losses, starting conditions, and draft and fuel different from those of the rating specifications. Unless the actual operation of the boiler based on the published capacity can be predicted with certainty it is desirable to make the selection on some other basis.

Method b involves ratings given to boilers by the Heating, Piping, and Air Conditioning Contractors National Association and the Steel Boiler Institute. In both cases the capacity stated is the net load which the boiler will carry when allowances have been made for piping losses. The HPACNA ratings allow a certain amount of radiation per square foot of grate area. Where the grate area is fixed as for a round boiler and the heating surface is dependent upon the number of sections installed, additional capacity is credited to a boiler when additional sections are installed. The SBI rates steel boilers in terms of net load per square foot of heating surface. The design and operation of a boiler have much to do with its actual performance. There is no assurance that boilers rated on physical characteristics alone will in all cases perform equally well.

Method c leads to satisfactory results when the test data may be applied to the installation under consideration. The chief difficulty with this particular scheme lies in the fact that curves showing the performance of heating boilers at various outputs and operating conditions are often difficult to obtain.

Method d is perhaps the best method of selection for hand-fired units. The engineer obtains, either as a result of his experience and judgment or from published data relative to conditions similar to those to be met, an estimate of the probable combustion rate and the operating efficiency that will be obtainable in the plant to be served. The combustion rate, of course, will be that possible with the chimney available when used with the type of boiler desired. The efficiency of operation will be dependent upon the kind, size, and heating value of the fuel, the method of firing, the character of the attendance given to the boiler, and the boiler itself as actually installed. The required grate area in square feet for solid-fuel hand-fired boilers is calculated as

$$G_a = \frac{\text{The total boiler load, Btu per hr}}{R_c \times F \times E_b} \tag{100}$$

where R_c = the combustion rate per sq ft, lb per hr.
F = heating value of the fuel, Btu per lb as fired.
E_b = overall boiler efficiency expressed as a decimal.

When the grate area is thus determined the choice between two boilers having the required grate area may be fixed by the selection of the one having the greater ratio of heating surface to grate area, as such a boiler will usually operate with lower flue-gas temperatures at the smoke outlet, provided that the heating surface is well arranged with respect to the gas flow and the water circulates freely in the boiler. This method differs from that of selecting a boiler on the basis that each square foot of grate area will supply a given amount of radiation in that it takes into consideration the combustion rate, the heating value of the fuel, and the efficiency of operation.

Heating-boiler combustion rates range from 3 to 20 lb per hr per sq ft of grate surface, depending upon the kind and size of the coal and the draft available.

191. Selection of Oil- and Gas-Fired Boilers. Combustion conditions are different in the furnaces of oil- and gas-fired boilers from those which burn solid fuels. Most oil and gas burners are operated using the "on-and-off" method. When the burner is "on," heat is released in the furnace at the maximum rate and unless the boiler has a sufficient amount of heat-absorbing surface the heat liberated may not be utilized to the fullest extent possible. During the off periods of burner operation care must be taken to prevent air from being drawn through the furnace and over the heating surfaces, for this would result in the loss of heat from the heated medium in the boiler to the air passing through the boiler and up the stack. This loss can be greatly reduced by an automatic damper in the base of the chimney or in the boiler-chimney connection which opens and allows air to flow into the chimney directly when the burner is not in operation. This by-pass action reduces the force of the draft. In solid-fuel-burning units, combustion proceeds at various rates as long as there is a fire in the furnace. With these units the maximum necessary heat release in the furnace occurs very much less frequently during the heating season than it does with oil- and gas-fired boilers, and thus there is less likelihood of unnecessary losses of heat in the flue gases.

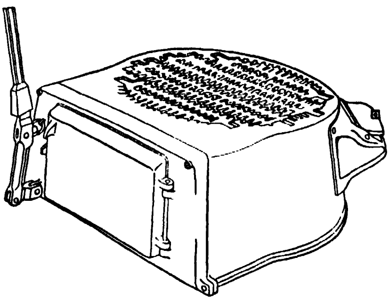


FIG. 117. Round cast-iron heating boiler ashpit section and grates.

Boilers using either gas or oil as a fuel should have ample amounts of heating surfaces and combustion spaces. Where oil is the fuel used the furnace should have a refractory lining in the combustion chamber.

In the selection of oil- and gas-fired boilers the manufacturer's ratings are the bases used for boiler selection. If the engineer believes the stated ratings to be excessive for the operating conditions existent then it is necessary for him to scale them down according to his experience and judgment.

192. Boiler Accessories and Auxiliaries. The grate bars of hand-fired boilers are of several types. A common type of base, forming the ashpit section, together with the grates for a round-type cast-iron sectional boiler, are shown by Fig. 117.

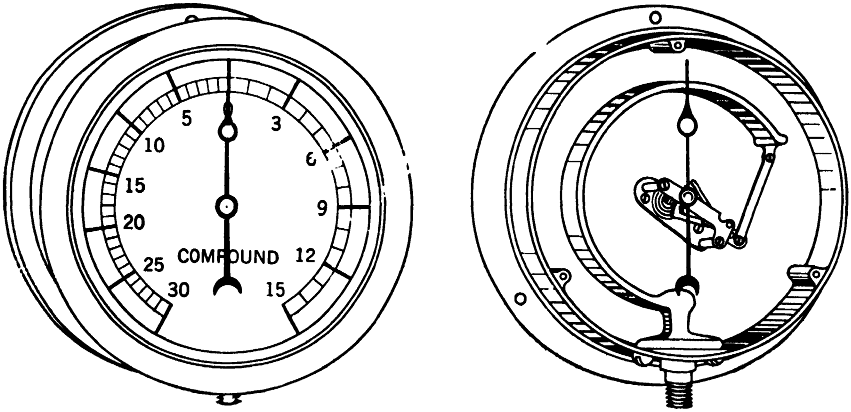


Fig. 118. Bourdon pressure gage.

Steam gages are usually of the Bourdon type as illustrated in Fig. 118. Where the boiler may be operated at pressures less than atmospheric the gage should be compound in that both pressures above and below atmospheric may be measured.

All steam boilers must have a safety valve which will discharge the maximum amount of steam that the boiler will generate when the boiler pressure exceeds a certain value. Safety valves are usually set to discharge steam at 5 to 10 psig above the usual pressure carried. With cast-iron boilers the safety valve must function at a pressure not to exceed 15 psig. Figure 119 illustrates a spring-loaded valve—the only type of valve that should be used.

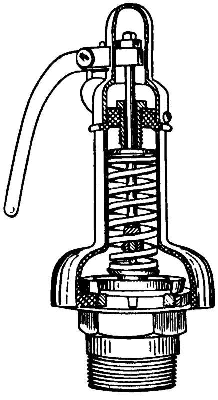


Fig. 119. Safety valve.

Damper regulators, though somewhat different for steam and hot-water service, usually consist of a bellows of corrugated metal which moves a lever to open and close the draft and check dampers. Steam regulators are actuated by steam pressure; hot-water regulators depend upon the volatilization of a liquid within the bellows to produce an operating pressure. Typical damper regulators are shown by Fig. 120.

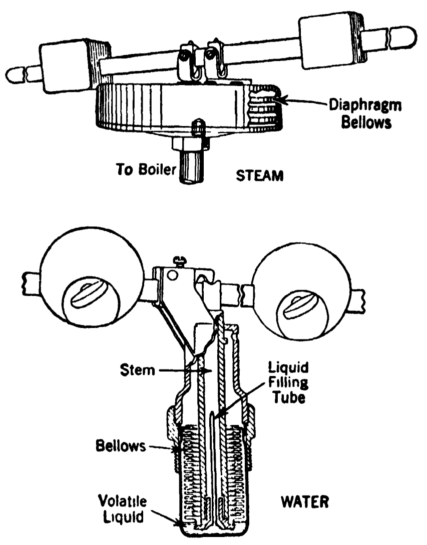


Fig. 120. Damper regulators.

Steam-heating boilers may or may not be equipped with a water column. One form of water column is shown by Fig. 121. The functions of the gage glass and the try cocks are to indicate the position of the water level in the boiler. The try cocks check the accuracy of the indications given by the gage-glass water levels. In the absence of a water column the gage glass and the try cocks are attached directly to the boiler.

Hot-water outlet temperatures are obtainable with a mercury thermometer as illustrated by Fig. 122. These thermometers are sometimes combined with the altitude gage, Fig. 123, used to indicate either the pressure of the water in the boiler or the height of the water level in the heating system.

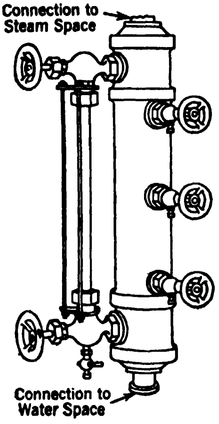


Fig. 121. Water column with gage glass and try cocks.

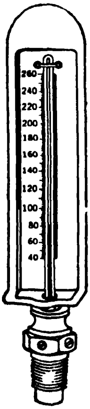


Fig. 122. Water-boiler thermometer.

Protection should be provided for steam-heating boilers in the form of equipment to stop the operation of automatic fuel-burning equipment and to allow the feeding of water to the unit whenever the level falls below a point where serious damage may happen. Figures 124 and 125 show the details of and the installation of a water feeder and a low-level cutout for a stoker-fired steam-heating boiler.

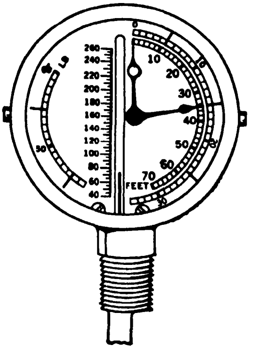
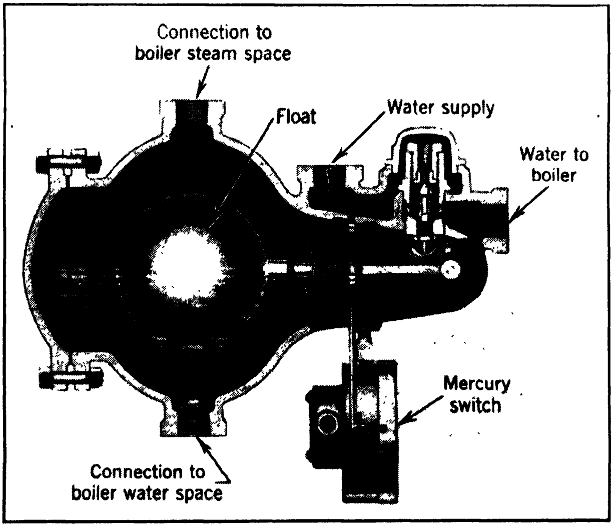


Fig. 123. Altitude gage and thermometer.

193. Water Heaters. Clean heated water is required in many buildings for domestic and other purposes. A number of operating reasons preclude the direct usage of the water of boilers of heating systems for the many and varied requirements of a supply of hot water. All water heaters are some form of a heat exchanger whereby the water heated is not contaminated as it passes through them. Often it is advisable to install an independently operated water heater which has as its source of heat either electricity or the combustion of either a solid, a liquid, or a gaseous fuel.


Connection to boiler steam space
Float
Water supply
Water to boiler
Mercury switch
Connection to boiler water space

(*McAlear Manufacturing Co.*)

Fig. 124. Water feeder with low-water alarm and automatic switch.

The following water-heating devices are employed in plants where boilers are installed: (1) furnace coils, (2) coils placed within the water space of a boiler, (3) units external to the boiler which secure their supply of heat from either steam or water of the boiler, and (4) heaters which utilize exhaust steam from engines, turbines, and other sources.

Coils placed in furnaces give variable water temperatures which are dependent upon the activity of the fire and the demands for heated water. A typical heater coil inserted within the water space of a boiler is that shown by Fig. 110. Figure 126 shows a type of water heater which may be connected into hot-water boilers and into steam boilers below the water line. Water from the boiler flows through the outer shell and heats the domestic water supply which flows through the bank of tubes of the heater. The tank heater, shown by Fig. 127, is for use with steam. The steam supply is regulated by a valve N which is under the control of a thermostatic element A placed in the heated water. The steam is condensed in the pipe coil, at the bottom of the tank, from which the condensate may be discharged to the

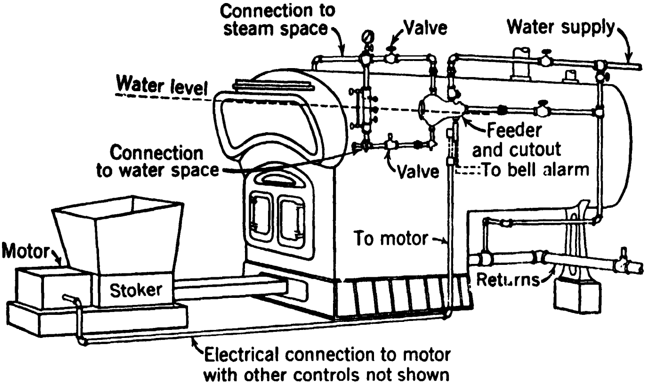

Connection to steam space
Valve
Water supply
Water level
Feeder and cutout
Connection to water space
Valve
To bell alarm
Motor
To motor
Stoker
Returns
Electrical connection to motor with other controls not shown

FIG. 125. Stoker-fired heating boiler with a water feeder.

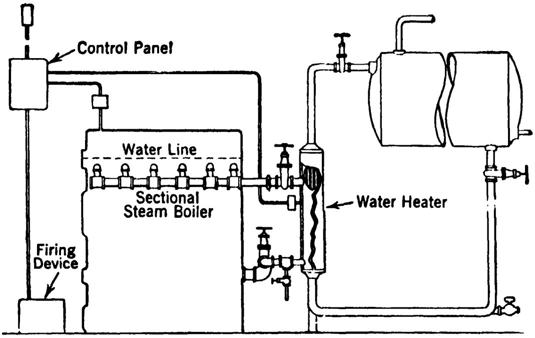


FIG. 126. Submerged water heater.

return line of the heating system. The tank heater of Fig. 127 may also be used without direct connection to a boiler where exhaust or other steam is available.

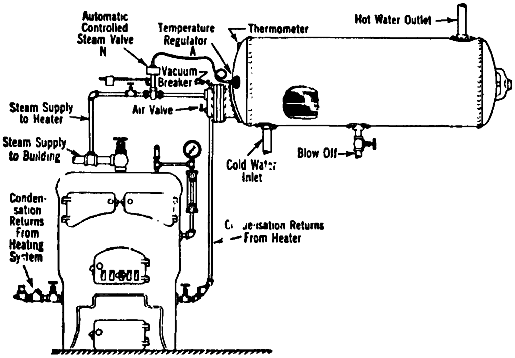


FIG. 127. Tank water heater.

The heat added to water passing through a heater per unit of time is equal to the product of the weight of water circulated and the change of the enthalpy of the liquid of the fluid.

PROBLEMS

1. A cast-iron heating boiler has a grate area of 3.97 sq ft. The gross I = B = R output is given as 295,000 Btu per hr and the net I = B = R rating as 134,000 Btu per hr. What factor has been allowed for piping losses and load pick-up? The boiler operated with an overall efficiency of 59 per cent. What combustion rate was necessary with coal having a heating value of 13,000 Btu per lb as fired?

2. An oil-fired boiler used 2.1 gal of number 3 furnace oil per hour to give a gross I = B = R output of 220,000 Btu per hr. What efficiency of operation was obtained? If the net I = B = R rating was 145,000 Btu per hr what total pick-up and piping-loss factor was allowed?

3. A steel heating boiler for hand-fired operation is rated at 8500 sq ft of equivalent steam radiation. What must be the amount of its heating surface? What amount of grate area is required?

4. A steel heating boiler for stoker firing is rated at 8500 sq ft of equivalent radiation when operated with steam. What amount of heating surface is required for the rating as stated? What should be the average height of the furnace if the plan area of the firebox is 20 sq ft?

5. A cast-iron heating boiler operated at 1.5 psig when the barometric pressure was 29.6 in. of mercury. During a period of 8 hr 8000 lb of steam of 0.99 quality were formed from water supplied at a temperature of 200 F. Find the boiler capacity in square feet of equivalent direct steam radiation and also boiler horsepower. Calculate the overall efficiency of operation if 6.5 lb of water were evaporated per each pound of 10,880 Btu per lb coal used during the test.

6. A hot-water boiler heated 9000 lb of water from 170 to 195 F. in one hour The weight of coal used per hour was 31 lb, and its heating value as fired was

11,200 Btu per lb. Find the overall efficiency of operation of the unit and the capacity developed in terms of standard hot-water radiation.

7. A building has heat losses as calculated for design purposes amounting to 2,350,000 Btu per hr. An allowance of 15 per cent is to be made for piping losses, and the air temperature is allowed to drop in the building at night time. The boilers may be expected to operate with an efficiency of 65 per cent and a combustion rate of 12 lb of coal per hr per sq ft of grate area. The coal has a heating value, as fired, of 11,500 Btu per lb. Find the required grate area of the necessary boilers, and select two steel units for the installation.

8. A residence with heat losses of 120,000 Btu per hr is to be heated by a hot-water system using a boiler burning number 2 furnace oil. Select an I = B = R rated boiler for this installation.

9. The water of a swimming pool, containing 60,000 gal, is to be heated from 52 to 70 F in 3 hr by steel boilers using coal of 12,300 Btu per lb calorific value as fired. The allowable combustion rate is 15 lb per hr per sq ft of grate area with a boiler efficiency of 68 per cent. Allow 10 per cent extra for piping losses. Find the required grate area of a boiler.

10. An oil-fired boiler is to heat a building having heat losses of 350,000 Btu per hr in the coldest weather. Domestic requirements of heated water are 200 gal. per hr raised in temperature from 55 to 160 F. Allow 20 per cent piping losses in the heating system and 5 per cent losses between the boiler and the water heater. How many gallons of number 3 fuel oil are required per hour if it can be utilized with 70 per cent efficiency?

CHAPTER 9

PIPE, TUBING, FITTINGS, COVERINGS, AND PIPING DETAILS

194. Pipe Materials. In heating and air-conditioning equipment both pipes and tubes are used for the confinement of and the conveyance of fluids such as water, steam, air, refrigerants, gas, etc. Materials used for pipes include wrought iron, mild steel, cast iron, aluminum, copper, and brass. Copper and brass are the most common tube materials although steel is used to some extent. For the most part non-ferrous materials resist corrosion better than those made from iron. Copper tubing has advantages in that it may be bent readily to make turns and offsets in lines and lengths of it can be easily fastened together with either solder or compression fittings thereby reducing labor costs in its installation. Thin-wall steel tubing has properties which allow it to be bent and welded together and is not subject to the corrosive action of such a material as ammonia when it is in contact with brass.

195. Commercial Pipe. Butt-welded and lap-welded steel and wrought-iron pipes are manufactured in sizes which are pertinent to their diameters and unless otherwise stated are furnished in random lengths up to 21 to 22 ft with threaded ends and couplings. Seamless pipes made by piercing and rolling pieces of steel may be had in relatively small sizes. For pipes up to 12 in. inclusive the sizes are based on the nominal inside diameters. Pipes designated as 14 in. and larger are listed on the basis of their outside diameters. All steel and wrought-iron pipes of the various sizes have external diameters which are standard for them irrespective of the thickness of the metal of their walls. This is necessary in manufacturing and also in order that the same thread-cutting equipment for each size, regardless of its wall thickness, may be used. The actual internal diameters of pipes vary from their listed sizes by amounts dependent upon their external diameters and the thickness of the wall metal.

The welding of pipe sections together and the use of welding-type fittings permits the use of thinner metal sections than when threading operations are necessary. At one time pipes were designated only by the terms standard weight, extra heavy, and double extra heavy according to their abilities to withstand internal pressures. Now the American Standard for Wrought-Iron and Wrought-steel Pipe, ASA, B36.10, lists the various commercial sizes of pipes under different schedules dependent upon their wall thickness. Schedules 30 and 40 of the American Standards Association conform most closely in dimensions to those classified as standard-weight pipes. Data pertinent to these schedules and standard-weight pipes are given in Table 68 as these are suitable for most uses with the low pressures of heating installations.

TABLE 68

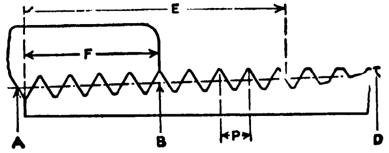
DIMENSIONS OF SCHEDULES 30 AND 40, WELDED STEEL PIPE FOR STEAM, AIR, GAS, AND WATER*

Nominal Size, In.	Inside Diameter, In.	External Diameter, In.	Nominal Thickness, In.	Transverse Areas			Length of Pipe Per Square Foot of External Surface, Ft	Length of Pipe Containing One Cubic Foot, Ft	Nominal Weight per Foot Plain Ends, Lb	Number of Threads per Inch of Screw
				Internal Sq In.	External, Sq In.	Metal, Sq In.				
$\frac{1}{8}$	0.269	0.405	**0.068**	0.057	0.129	0.072	9.431	2533.775	0.244	27
$\frac{1}{4}$	0.364	0.540	**0.088**	0.104	0.229	0.125	7.073	1383.789	0.424	18
$\frac{3}{8}$	0.493	0.675	**0.091**	0.191	0.358	0.167	5.658	754.360	0.567	18
$\frac{1}{2}$	0.622	0.840	**0.109**	0.304	0.554	0.250	4.547	473.906	0.850	14
$\frac{3}{4}$	0.824	1.050	**0.113**	0.533	0.866	0.333	3.637	270.034	1.130	14
1	1.049	1.315	**0.133**	0.864	1.358	0.494	2.904	166.618	1.678	11$\frac{1}{2}$
1$\frac{1}{4}$	1.380	1.660	**0,140**	1.495	2.164	0.669	2.301	96.275	2.272	11$\frac{1}{2}$
1$\frac{1}{2}$	1.610	1.900	**0.145**	2.036	2.835	0.799	2.010	70.733	2 717	11$\frac{1}{2}$
2	2.067	2.375	**0.154**	3.355	4.430	1.075	1.608	42.913	3.652	11$\frac{1}{2}$
2$\frac{1}{2}$	2.469	2.875	**0,203**	4.788	6.492	1.704	1.328	30.077	5.793	8
3	3.068	3.500	**0,216**	7.393	9.621	2.228	1.091	19.479	7.575	8
3$\frac{1}{2}$	3.548	4.000	**0.226**	9.886	12.566	2.680	0.954	14.565	9.109	8
4	4.026	4.500	**0.237**	12.730	15.904	3.174	0.848	11.312	10.790	8
5	5.047	5 563	**0.258**	20.006	24.306	4.300	0.686	7.198	14.617	8
6	6.065	6.625	**0.280**	28.891	34.472	5.581	0.576	4.984	18.974	8
8†	8.071	8.625	0.277	51.161	58.426	7.265	0.443	2 815	24.696	8
8	7.948	8.625	**0.322**	50.027	58.426	8.399	0.443	2.878	28 554	8
10†	10.136	10.750	0.307	80.691	90.763	10.072	0.355	1.785	34 240	8
10	10.020	10.750	**0.365**	78.855	90.763	11.908	0.355	1.826	40.483	8
12†	12.090	12.750	0.330	114.800	127.676	12.876	0.299	1.254	43.773	8
12	12.000	12.750	0.375	113.097	127.676	14.579	0.299	1.273	49.562	8

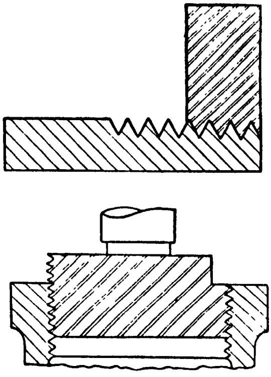
* Nominal thickness given in bold-faced type is identical with that for schedule 40.

† Thickness same as for schedule 30.

Brass and copper pipes from $\frac{1}{8}$ to 12 in. are also listed with outside diameters equal to those given in Table 68. Such pipes can be used with fittings conforming in size to those suitable for steel and wrought-iron standard pipe. The internal diameters of brass pipe for the most part differ slightly from those of standard steel pipe. Aluminum pipe with ips, iron pipe size, welding fittings is finding use in non-corrosive installations in chemical, food soap, petroleum, and cosmetic plants. The cost is less than that of either brass or copper pipe.



A = pitch diameter of thread at end of pipe.
B = pitch diameter of thread at gaging notch; also, pitch diameter of straight pipe thread.
D = outside diameter of pipe.
E = length of effective thread.
F = normal engagement by hand between external and internal threads.
P = pitch of thread.
$A = D - (0.050D + 1.1)P$
$B = A + 0.0625F$
$E = (0.80D + 6.8)P$
Depth of thread $= 0.80P$
Taper = $\frac{3}{4}$ in. per ft.



Plug gage to go in until notch is flush with face. Ring gage to go in until flush with end of pipe.
Allowable variations: one turn, plus or minus.

Fig. 128. American standard pipe thread.

196. Pipe Threads. Pipe threads and the tappings of the various threaded fittings for commercial pipe are standardized and have a taper of 1 in 16 with the longitudinal axis of the pipe or the tapping. The threads are V shaped with a 60-deg angle between them and are flattened at their tops at the larger end of the taper, Fig. 128. The function of the taper is to produce a very tight joint when the threads draw together the parts to be joined.

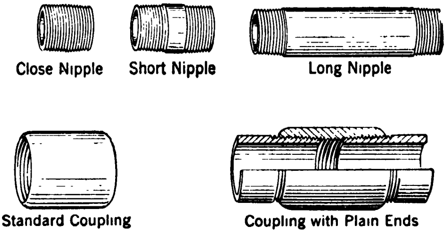


Fig. 129. Pipe nipples and couplings.

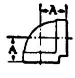
197. Pipe Fittings for Standard Steel and Wrought-Iron Pipe. These parts, for joining lengths of pipe together and providing outlet connections, reductions of pipe size, etc., are made in two forms. The screw type of fitting is usual for pipes less than 4 in. in diameter and the flanged type of fitting for pipes 4 in. in diameter and larger. There are exceptions to this last statement.

Materials for fittings include cast iron, malleable iron, steel, and brass, the choice depending on the service and pressure. The weight of fittings should conform to the weight of pipe; thus standard fittings

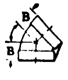
TABLE 69

DIMENSIONS OF STANDARD CAST-IRON SCREWED PIPE FITTINGS

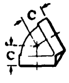
Working pressure 125 psig



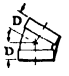
90° Elbow



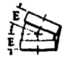
45° Elbow



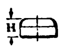
60° Elbow



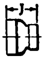
30° Elbow



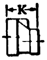
22½° Elbow



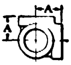
Cap



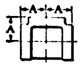
Reducer



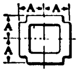
Eccentric Reducer



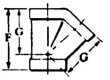
Side Outlet Elbow



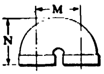
Tee



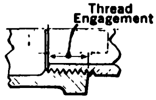
Cross



45° Y-Bend



Return Bend



(Crane Co.)

Size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*J*	*K*	*H*
1/4	13/16	3/4								
3/8	15/16	13/16								
1/2	1 1/8	7/8	1	13/16	3/4	2 1/2	1 7/8			
3/4	1 5/16	1	1 1/16	15/16	7/8	3	2 1/4	1 1/2		
1	1 1/2	1 1/8	1 1/4	1	1	3 1/2	2 3/4	1 7/8*		
1 1/4	1 3/4	1 5/16	1 7/16	1 3/16	1 1/8	4 1/4	3 1/4	2 1/8	2 1/8	
1 1/2	1 15/16	1 7/16	1 5/8	1 5/16	1 1/4	4 7/8	3 13/16	2 1/4	2 1/4	
2	2 1/4	1 11/16	1 7/8	1 3/4	1 7/16	5 3/4	4 1/2	2 7/16	2 7/16	
2 1/2	2 11/16	1 15/16	2 3/16	1 13/16	1 9/16	6 3/4	5 3/16	2 5/8	2 11/16	
3	3 1/8	2 3/16	2 1/2	1 13/16	1 3/4	7 7/8	6 1/8	2 7/8	2 15/16	
3 1/2	3 7/16	2 3/8				8 7/8	6 7/8	3 1/8	3 1/8	
4	3 3/4	2 5/8	3	2 3/8	2 1/16	9 3/4	7 5/8	3 3/8	3 3/8	2 1/16
5	4 1/2	3 1/16	3 1/2	2 5/8	2 1/4	11 5/8	9 1/4	3 7/8	3 7/8	2 3/8
6	5 1/8	3 7/16	4 1/16	2 7/8	2 7/16	13 7/16	10 3/4	4 3/8	4 3/8	2 5/8
8	6 9/16	4 1/4				16 15/16	13 5/8	5 1/4	5 1/4	3 1/8
10	8 1/16	5 3/16				20 11/16	16 3/4	6 3/16		3 5/8
12	9 1/2	6				24 1/8	19 5/8	7 1/8		4 1/4

Return Bends

Size	*M*	*N*	Size	*M*	*N*	Size	*M*	*N*	Size	*M*	*N*
Close Pattern			Open Pattern			Wide Pattern			Wide Pattern		
1/2	1 1/4	1 23/32	3/4	1 7/8	2 7/32	1	3	3	1 1/2	4 7/8	4 7/16
3/4	1 1/2	2 1/32	1	2 1/2	2 11/16	1	4	3 1/2	1 1/2	6	5
1	1 3/4	2 3/8	1 1/4	3	3 9/32	1	5	4	1 1/2	8	6
1 1/4	2 1/4	2 29/32	1 1/2	3 1/2	3 3/4	1	6	4 1/2	2	4 7/8	4 3/4
1 1/2	2 1/2	3 1/4	2	4 1/2	4 19/32	1	8	5 1/2	2	6	5 5/16
2	3 1/4	3 31/32	2 1/2	5 1/2	5 7/16	1 1/4	4	3 3/4	2	7	5 13/16
2 1/2	3 3/4	4 9/16	3	6 1/2	6 5/16	1 1/4	6	4 3/4	2	8	6 5/8
3	4 1/4	5 3/16	4	7 1/2	7 9/16	1 1/4	8	5 3/4	4	11	9 5/16
4	6	6 13/16									

* 1 × 1/2-in. reducers are 1 11/16 in. end to end

are used with standard weights of pipe for pressures up to 125 psig and extra-heavy fittings with extra-heavy pipe for pressures up to 250 lb. Extra-heavy fittings and extra-heavy pipe are not required in low-pressure steam heating. Close and shoulder nipples and ordinary couplings, Fig. 129, are made either of steel or wrought iron. The names of common fittings and their appearances are indicated by the various details of Fig. 130. Table 69 gives data relative to important dimensions of several common cast-iron screw fittings.

The size of fittings is always designated in accordance with the pipe with which they are to be used The tappings of fittings, such as elbows, tees, and couplings, may be uniform in size; or one or more of the tappings, depending upon the fitting, may be smaller, when the term reducing fitting is applied. If the side outlet of a tee is for 1-in. pipe and the other two tappings are for $1\frac{1}{2}$-in. and $1\frac{1}{4}$-in. pipe, respectively, the fitting is a reducing tee, and the size designation is $1\frac{1}{2} \times 1\frac{1}{4} \times 1$ in. The side-outlet size of a tee is always given last.

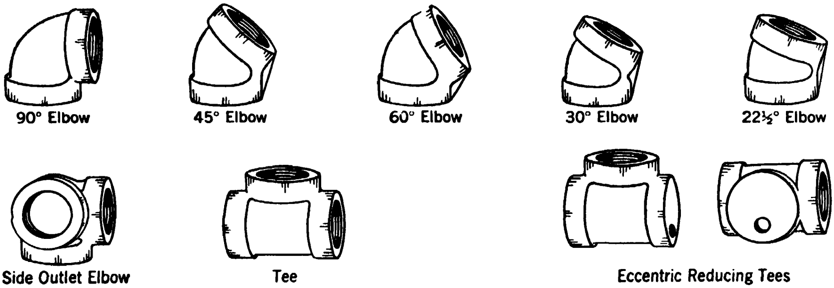
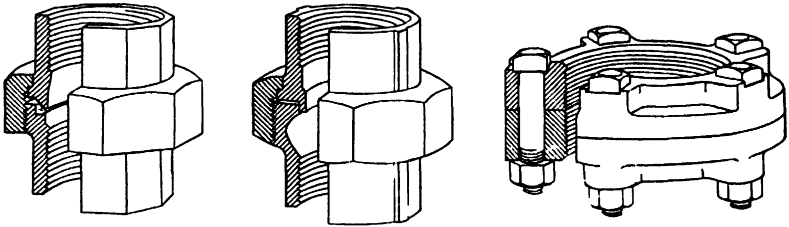


FIG. 130. Screwed pipe fittings.

Provision must always be made when two immobile objects are fastened together, by pipe connections, for the final joining of the last two sections of pipe and for the easy disconnection of pipes at various locations in a system when repairs are necessary. With small pipes this can be accomplished by means of couplings having a right-hand thread in one end and a left-hand thread in the other end. These couplings usually have a distinguishing mark such as longitudinal ridges on their exterior. Unions or companion flanges are more satisfactory than fittings with right- and left-hand threads.

Unions, Fig. 131, are of two different forms, nut unions and flange unions. Nut unions consist of three parts which are the two parts attached to the pipe ends by threads and the threaded nut which is used to draw the seat parts together. Depending upon their construction nut unions may or may not require a gasket where their seats fit together. A flange union consists of two parts which are attached by threads to the pipe joined together. A gasket is placed between the flanges which are drawn together by bolts with nuts, or the flange union may have ground seats which do not require a gasket. Nut unions are made in all pipe sizes up to 4 in. inclusive. For pipe sizes above 2 in. flange unions are generally preferred.

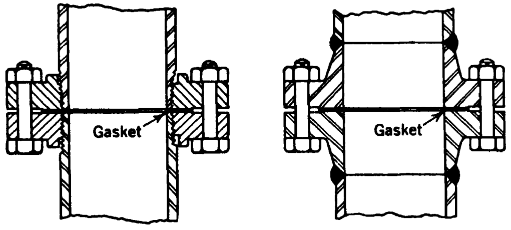


Gem Female Standard Flange, Plain Face

(*Crane Co.*)

FIG. 131. Pipe unions.

Flanged fittings are desirable for pipe sizes of 4 in. and larger and are used in connection with companion flanges attached to the pipe ends. Examples of pipes joined together with flanges are shown by Fig. 132. Companion flanges are available in either standard weight or extra heavy and are drawn together with bolts. Flanged joints usually require a gasket of some material between the flange faces inside of the bolt circle. Flanged joints serve as unions and are of great convenience in assembling and taking down pipe lines. Typical flanged fittings suitable for pressures up to 125 psig are shown by the figures and data of Table 70.



Screwed Welded Neck

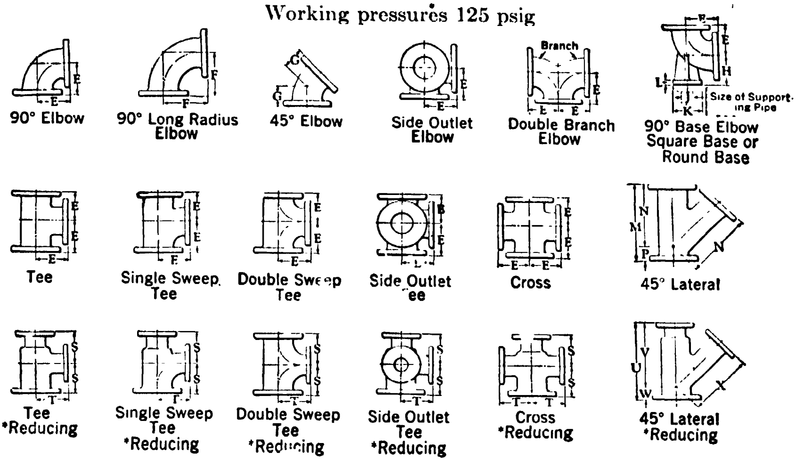
FIG. 132. Flanged pipe joints.

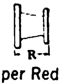
198. Valves. The stop valves used in heating pipe lines may be angle, globe, or gate types, Figs. 133 and 134, made of brass or cast iron. Gate valves are usually made only in the straight-away pattern, although an angle-type has been manufactured; globe-valve construction may be either the straight-away or the angle pattern. Gate valves are desirable as they offer the least amount of obstruction to

TABLE 70

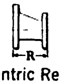
Dimensions of Standard Cast-Iron Flanged Pipe Fittings

Working pressures 125 psig

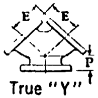




Taper Reducer



Eccentric Reducer



True "Y"

* *Dimensions of reducing fittings.* The center-to-face dimensions of reducing elbows, tees, crosses, and 45° laterals are the same as for the straight size corresponding to the size of the largest opening, *with the following exception.*

Exception. In sizes 18 in. and larger, if the outlet of a reducing tee, cross, or 45° lateral is the size given in the table below, or is smaller, use the "short-body" dimensions shown below.

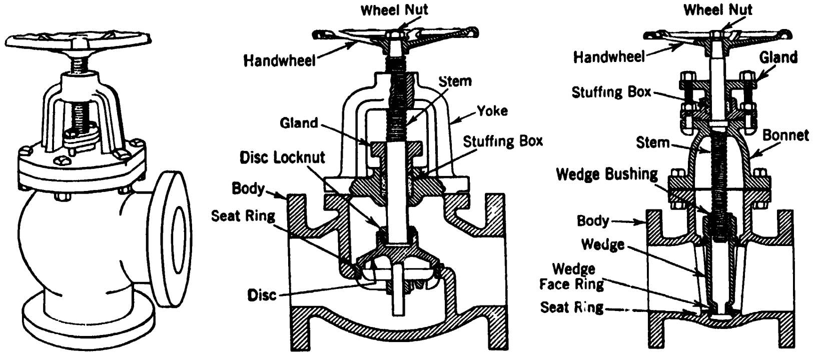
The face-to-face dimension of reducers is governed by the size of the larger opening, regardless of the size of the smaller opening.

Size	*E*	*EE*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*N*	*P*	*R*
1	3½	7	5	1¾	3½	¾	3½	7/16	7½	5½	1¼	
1¼	3¾	7½	5½	2	3⅝	¾	3½	7/16	8	6¼	1¾	
1½	4	8	6	2¼	3¾	1	4¼	7/16	9	7	2	
2	4½	9	6½	2½	4⅛	1¼	4⅝	½	10½	8	2½	5
2½	5	10	7	3	4½	1¼	4⅝	½	12	9½	2½	5½
3	5½	11	7¾	3	4⅞	1½	5	9/16	13	10	3	6
3½	6	12	8½	3½	5¼	1½	5	9/16	14½	11½	3	6½
4	6½	13	9	4	5½	2	6	5/8	15	12	3	7
5	7½	15	10¼	4½	6¼	2½	7	11/16	17	13½	3½	8
6	8	16	11½	5	7	2½	7	11/16	18	14½	3½	9
8	9	18	14	5½	8⅜	4	9	15/16	22	17½	4½	11
10	11	22	16½	6½	9¾	4	9	15/16	25½	20½	5	12
12	12	24	19	7½	11¼	6	11	1	30	24½	5½	14
14	14	28	21½	7½	12½	6	11	1	33	27	6	16
16	15	30	24	8	13¾	6	11	1	36½	30	6½	18
18	16½	33	26½	8½	15	8	13½	1⅛	39	32	7	19
20	18	36	29	9½	16	8	13½	1⅛	43	35	8	20
24	22	44	34	11	18½	8	13½	1⅛	49½	40½	9	24
30	25	50	41½	15					59	49	10	30
36	28	56	49	18								36
42	31	62	56½	21								
48	34	68	64	24								

Short-Body Reducing Fittings

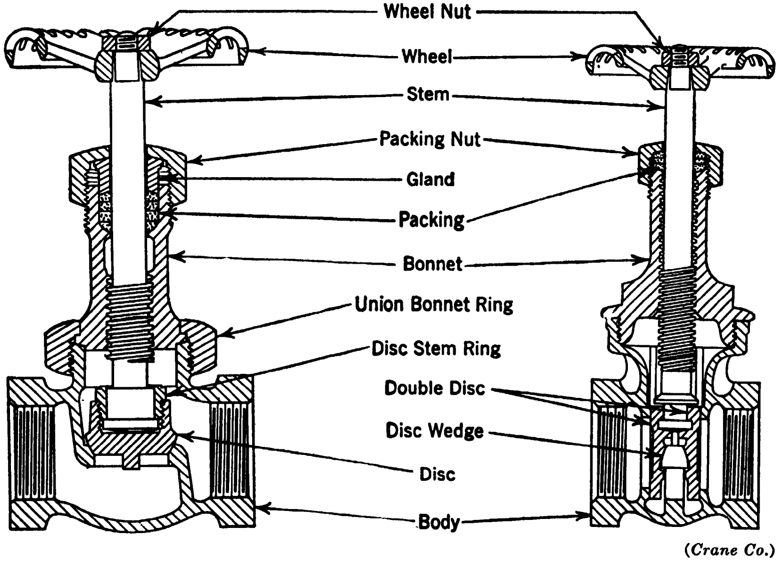
Size	Tees and Crosses				45° Laterals				
	Size of Outlet and Smaller	*S*	*SS*	*T*	Size of Outlet and Smaller	*U*	*V*	*W*	*X*
18	12	13	26	15½	8	26	25	1	27½
20	14	14	28	17	10	28	27	1	29½
24	16	15	30	19	12	32	31½	½	34½
30	20	18	36	23	14	39	39	0	42
36	24	20	40	26					

the flow of steam and water. Globe valves secure a throttling action by partially closing the valve. The use of a globe valve, with its stem vertical, in a horizontal steam line is not recommended as it creates an obstruction in the pipe which will cause an accumulation of water to choke the flow of steam. Furthermore, complete drainage of the line is impossible. Brass valves come in sizes of 3 in. and less.



Angle Globe Gate

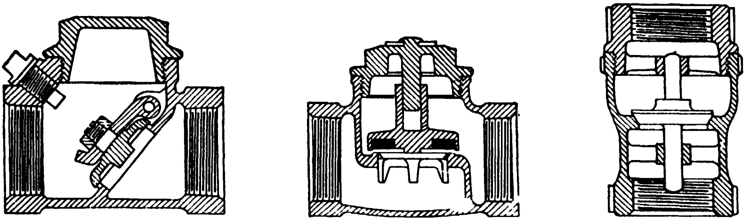
FIG. 133. Stop valves.



(*Crane Co.*)

FIG. 134. Small globe and gate valves.

For larger pipes most valves have cast-iron bodies. Some cast-iron valves have brass inserts in the seats, discs, and gates. These inserts



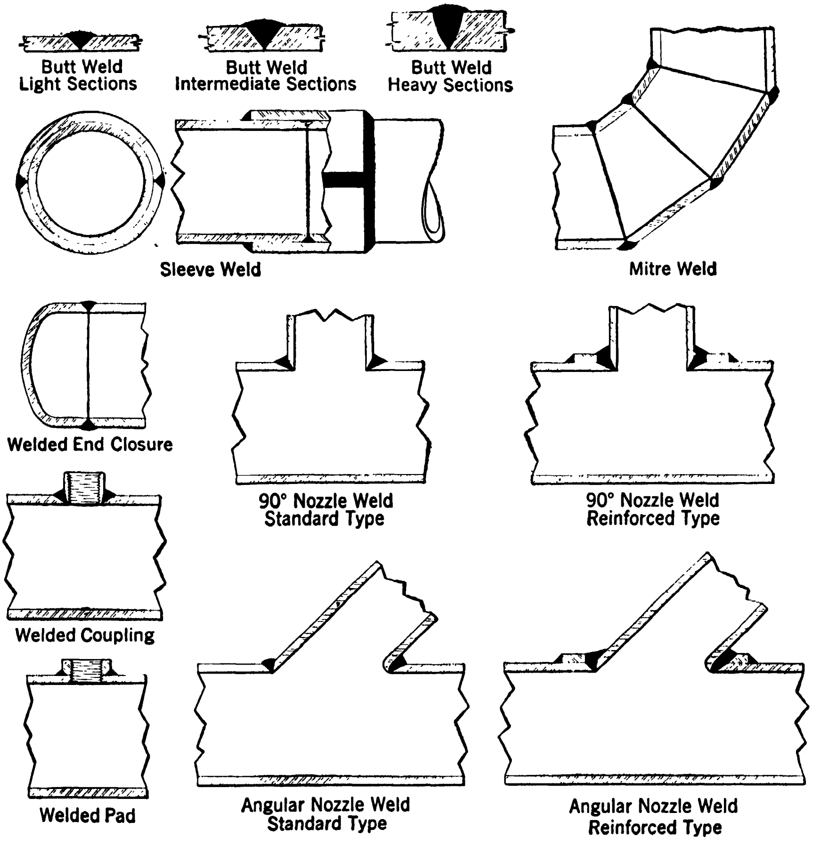
Swing

Horizontal

Vertical

(*Crane Co.*)

FIG. 135. Cross sections of check valves.



Butt Weld Light Sections
Butt Weld Intermediate Sections
Butt Weld Heavy Sections
Sleeve Weld
Mitre Weld
Welded End Closure
90° Nozzle Weld Standard Type
90° Nozzle Weld Reinforced Type
Welded Coupling
Welded Pad
Angular Nozzle Weld Standard Type
Angular Nozzle Weld Reinforced Type

(*Crane Co.*)

FIG. 136. Pipe welds.

are corrosion resisting and are an aid in keeping the valve tight against leakage through it.

Gate valves may have either a rising or a non-rising stem; globe valves always have a rising stem. The rising stem requires more space but has the advantage of giving positive indication of the valve opening. All the valves of Figs. 133 and 134 require packing of some kind within the stuffing boxes of the bonnets and around the valve stems to prevent leakage of the fluid passing through them.

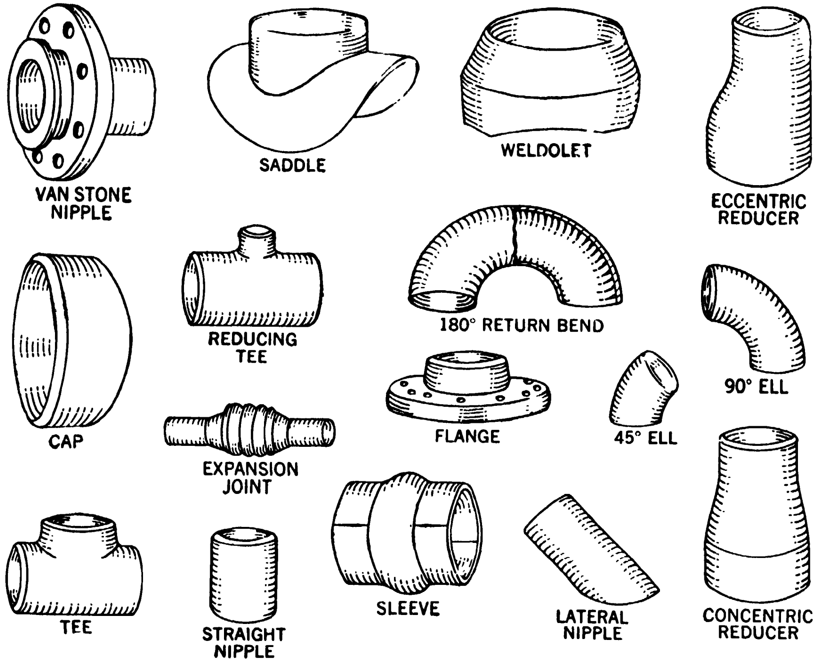


FIG. 137. Welding fittings.

Check valves serve to prevent backward flow in a pipe line and are designed for horizontal and vertical lines. The check valves of Fig. 135 are for horizontal and vertical pipes. The bodies of check valves are commonly made of brass in the smaller sizes and of cast iron in the larger ones.

199. Welded Pipe-Joints. Considerable savings in material and labor costs can often be made by welding together pipe sections and fittings. Also special connections, in the field, can be improvised by the welder. Fusion welding, either electrical or by the use of a gas flame, is not economical with pipes smaller than 2 in. in diameter.

The parts to be joined have the edges of the material scarfed or beveled to a suitable angle and are butted together; the metal deposit from the welding rod is built up in the groove thus formed on the outside as shown by Fig. 136. Elbows may be made of short lengths of pipe, or they may be purchased fabricated as a seamless tube. Valves and flanges may also be obtained for welding into steam or other lines. Examples of elbows and tees formed of pipe by welding are shown by Fig. 136. Commercial welding fittings are depicted by the examples of Fig. 137.

200. Steel Tubing. The material used in ferrous tubing may be either of carbon or alloy steel. Generally the listed sizes of steel tubes are based on their external diameters which have tolerances of deviation of the actual from the nominal size of only a few hundredths of an inch. The wall thickness of steel tubes vary with their external diameters and the pressures which they must withstand. Tubes of small size may be readily bent, and lengths of them, having thin walls, may be joined together by welding. Applications of such pipe construction are in condensers of various kinds, flues of fire-tube boilers, water passages of water-tube boilers, and finned- or extended-surface areas of either cooling or heating coils. Steel tubing is available in duplex-wall construction, Fig. 138, for use where corrosion difficulties may be encountered. Such construction is obtained by drawing a tube of copper or copper alloy either inside or outside of the steel one.

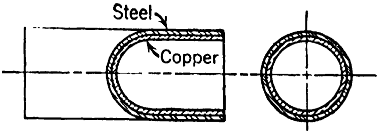


FIG. 138. Duplex steel and copper tube.

201. Copper Tubing. Seamless tubing for heating, plumbing, and air-conditioning installations is made of either soft and ductile or hard-drawn copper. Copper tubing is standardized as Type K, L, or M depending upon the thickness of the wall material. For a given tube size, Type K has the thickest and type M the thinnest wall.

Data relative to the dimensions and the weights of copper tubing are given by Table 71.

All Type-M copper tubing is hard drawn or has a hard temper. Types K and L may be either hard drawn or annealed. In any case the bursting strength of a given type is dependent upon the physical characteristics of the copper. Standard lengths of copper tubing are 20 ft, but soft-drawn Types K and L of $1\frac{1}{4}$-in. diameter or less are furnished in coiled lengths of 30, 45, and 60 ft.

Type-K tubing is suitable for underground water-service lines, plumbing, heating, oil lines, and some air-conditioning installations.

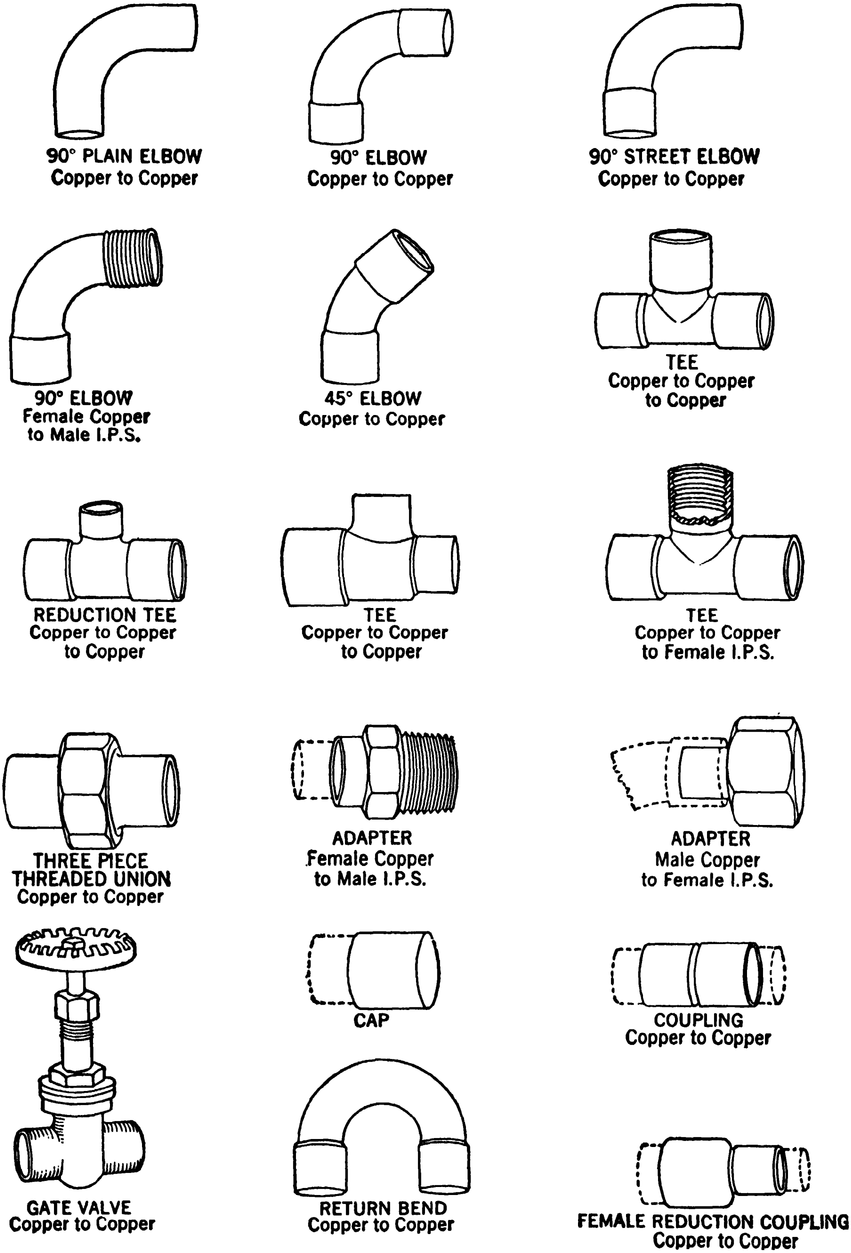


Fig. 139. Copper-tube fittings.

TABLE 71

Copper Tubing for General Plumbing and Heating

Nominal Size, In.	Outside Diameter, In.	Type K for Underground Service, General Plumbing and Heating Installations under Severe Conditions		Type L for General Plumbing and Heating Installations		Type M for General Plumbing and Heating Installations, Straight Hard Lengths Only	
		Wall Thickness, In.	Weight per Foot, Lb	Wall Thickness, In.	Weight per Foot, Lb	Wall Thickness, In.	Weight per Foot, Lb
1/4	0.375	0.032	0.133	0.030	0.126	0.025	0.106
3/8	0.500	0.049	0.269	0.035	0.198	0.025	0.144
1/2	0.625	0.049	0.344	0.040	0.285	0.028	0.203
5/8	0.750	0.049	0.418	0.042	0.362		
3/4	0.875	0.065	0.641	0.045	0.455	0.032	0.328
1	1.125	0.065	0.839	0.050	0.655	0.035	0.464
1 1/4	1.375	0.065	1.04	0.055	0.884	0.042	0.681
1 1/2	1.625	0.072	1.36	0.060	1.14	0.049	0.940
2	2.125	0.083	2.06	0.070	1.75	0.058	1.46
2 1/2	2.625	0.095	2.92	0.080	2.48	0.065	2.03
3	3.125	0.109	4.00	0.090	3.33	0.072	2.68
3 1/2	3.625	0.120	5.12	0.100	4.29	0.083	3.58
4	4.125	0.134	6.51	0.110	5.38	0.095	4.66
5	5.125	0.160	9.67	0.125	7.61	0.109	6.66
6	6.125	0.192	13.87	0.140	10.20	0.122	8.91
8	8.125	0.271	25.90	0.200	19.29	0.170	16.46

Types K and L are adapted to the use of compression and flared fittings as well as solder in the smaller sizes and solder in the larger sizes. Type-M tubing, furnished with thin walls and a hard temper, is used with solder fittings only.

Solder fittings such as tees, elbows, reducers, valves, adapters, caps, and unions, Fig. 139, are sold under various trade names. Figure 140 shows the process of making a joint with a solder fitting. Copper tubes can be joined to the fittings of steel pipes by means of adapter units having both threaded and solder, flared, or ferrule connections.

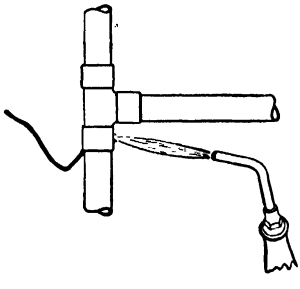
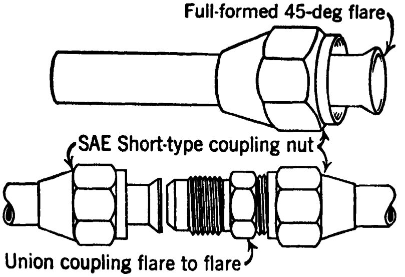


Fig. 140. Soldered joint.

Small tubes may be fastened to their fittings by flared joints as shown by Fig. 141. Figure 142 shows the use of a ferrule or sleeve in a joint. Flared joints are made by cutting square ends on the tubes to be joined. A connector nut is then slipped over each tube end, and a seat is formed at the tube end with a special tool. The parts are then drawn together by the connector nut as shown in Fig. 141. When a ferrule is used, Fig. 142, tube flaring is not necessary as the nut compresses the ferrule to make a tight joint between it and the tube and also where the ferrule bears on 45-deg seats of the parts joined.



(*American Brass Co.*)

FIG. 141. A full-formed copper-tube flare and the completed joint ready for assembly.

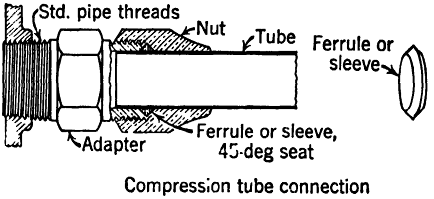


FIG. 142. Tube connection with adapter and ferrule or sleeve.

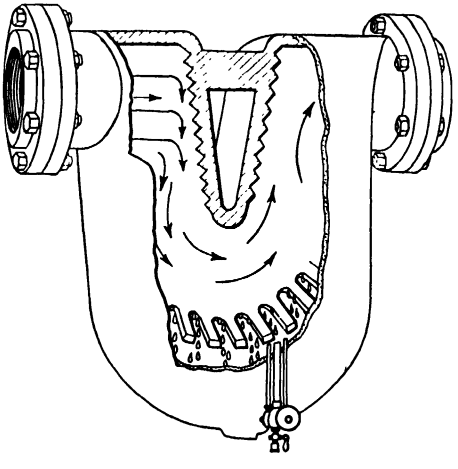


FIG. 143. Steam separator.

202. Steam Separators and Traps. A separator is a piece of equipment for removing water particles from steam as it flows in a pipe line or for removing oil from the steam exhausted by engines. The separator illustrated by Fig. 143 causes a change of direction of flow of steam passing through it to aid in the removal of the water particles. The water collects in the bottom chamber.

Traps are devices which permit the removal of collected water without the loss of steam from heaters, radiators, separators, and steam-pipe lines. Traps may be classified as float, bucket, bowl, and thermostatic. The float trap of Fig. 144*a* operates as water entering at *I* accumulates and causes the float *C* to lift to open valve *F*. The steam pressure then forces water through the outlet *H* until the float is lowered to close valve *F*. The trap of Fig. 144*b* has a thermostatic

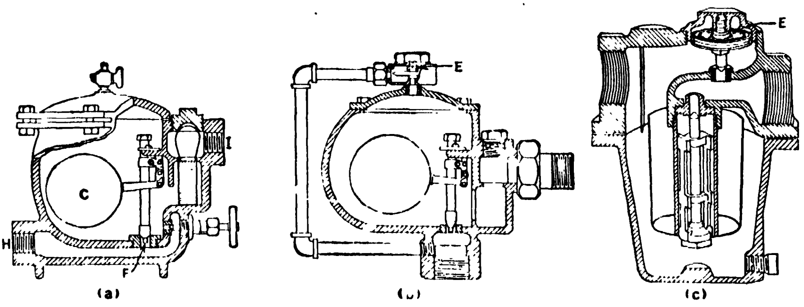
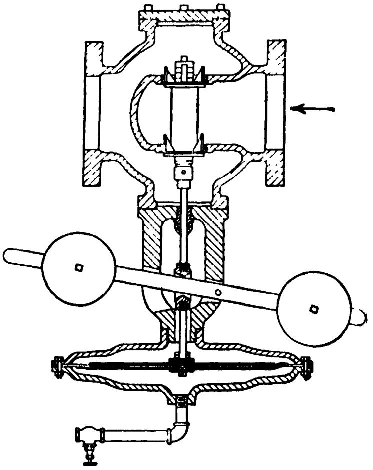


Fig. 144. (*a*) Float trap. (*b*) Float and thermostatic drip trap. (*c*) Bucket and thermostatic drip trap.

element at *E* to allow the discharge of air. The bucket trap shown in section by Fig. 144*c* has a bucket which floats and holds the valve against the seat of the outlet until the water overflows into the bucket. When this occurs the bucket sinks, and steam pressure causes a discharge of water through the outlet until the bucket rises again to close the discharge opening. This trap also has a thermostatic element *E* which permits the discharge of air. Thermostatic radiator traps are described in Chap. 10.

203. Pressure-Reducing Valves. When steam is drawn from high-pressure lines to heat buildings its pressure must be reduced to the working pressure of the system. This is accomplished by means of a pressure-reducing valve, one form of which is shown in Fig. 145. A pressure equal to that desired in the heating supply piping is maintained beneath the diaphragm by means of the balance-pipe connection between the low-pressure main and the valve. This pressure can be controlled by the positions of the weights on the lever which



(*Illinois Engineering Co.*)

Fig. 145. Pressure-reducing valve.

shifts the spindle or stem attached to the discs controlling the openings of the valve through which steam may flow from the high-pressure to the low-pressure piping. The valve should be installed with a by-pass connection so that service by means of hand operation may be maintained when the pressure-reducing valve is out of service.

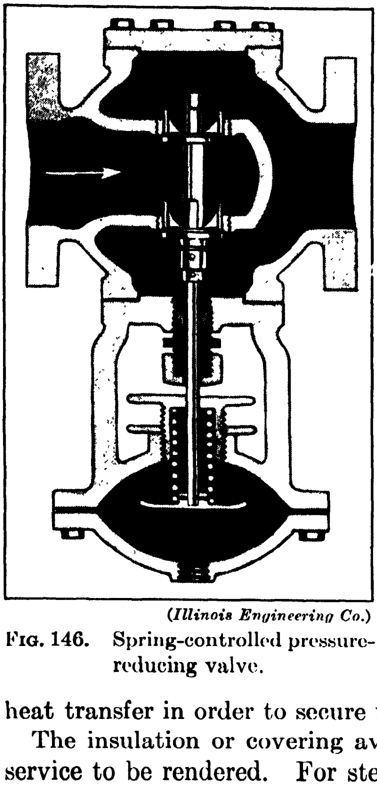

(*Illinois Engineering Co.*)

FIG. 146. Spring-controlled pressure-reducing valve.

Another form of pressure-reducing valve, Fig. 146, utilizes a coil spring, the compression of which can be changed by an adjusting screw, as an alternative to the lever and weight arrangement of the valve shown by Fig. 145.

204. Pipe Insulation. Pipes conveying a medium at a temperature greater or less than that of the air surrounding them have a flow of heat through their walls, causing the medium to be either cooled or heated as the case may be. Where the medium is steam, hot water, chilled water, or cold brine it is desirable to partially minimize this heat transfer in order to secure reduced operating costs.

The insulation or covering available for pipe lines varies with the service to be rendered. For steam and hot-water lines coverings are made of magnesia, asbestos, diatomaceous earths, felts, and metal foils. Lines conveying cold liquids are covered with molded cork or sometimes with hair felt. To be effective the insulation must always be kept dry. With the exceptions of metal foils, cork, and hair felt the insulators mentioned are combined with other materials which give strength and rigidity to the covering.

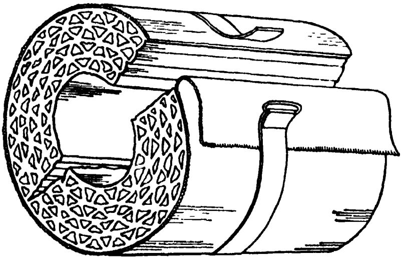


FIG. 147. Cellular pipe covering.

Magnesium carbonate mixed with asbestos fibers molded into blocks for flat surfaces or into split circular sections for pipes is an efficient insulator and is widely known as "85 per cent magnesia." Asbestos mixed with a binder is rolled into sheets to produce cellular coverings as shown by Fig. 147. Results of tests[1] of several commercial pipe coverings are shown by Fig. 148. The heat losses given are based on each square foot of the external surface of the pipe covered and are not per square foot of external surface of the covering.

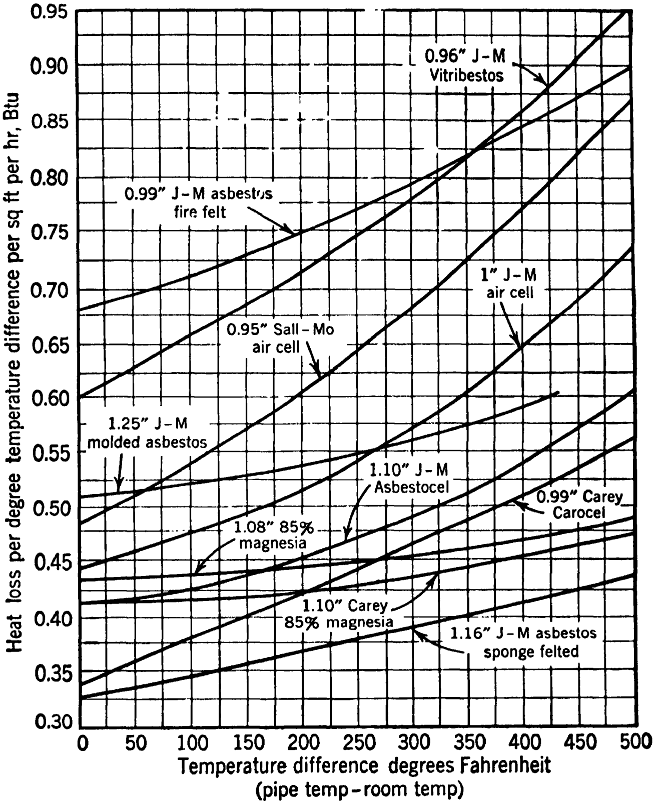


FIG. 148. Heat losses from covered pipe.

Pipe fittings of various sorts are enclosed with special molded coverings made to fit them or with an insulating cement containing asbestos fibers. This cement is applied with a trowel, and the finished covering is wrapped with canvas to give additional strength and protection to the covering. Boilers may be insulated with magnesia blocks and an external coat of insulating cement and canvas, or they may be covered with 2 in. of plastic cement reinforced with wire netting, the final outside surface being finished with a thickness of canvas. Insulation of jacketed boilers is described in Chap. 8.

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[1] "The Heat Insulating Properties of Commercial Steam Pipe Coverings," by L. B. McMillan, *ASME Trans.*, Vol 37, 1915.

The efficiency of pipe covering is expressed as one minus the ratio of the heat actually lost from the pipe through the covering to that which would have been lost under the same conditions of inside-steam and outside-air temperature from one square foot of uncovered pipe surface. Data for the heat losses of bare pipes are shown by Fig. 149. Depending upon local conditions there is an economical limit in the use of pipe insulations with respect to their materials and the thickness of the coverings.

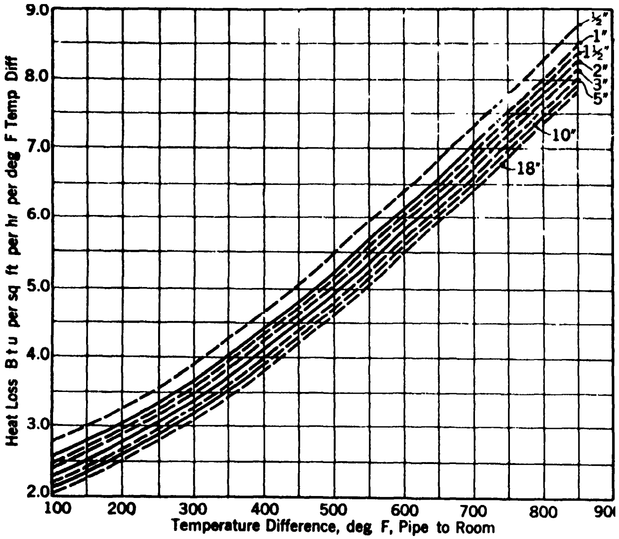

FIG. 149. Heat losses from horizontal bare steel pipe.

Copper tubing together with its fittings has a bright finish when new that tends to prevent losses of heat from its surfaces. This bright finish can be preserved by the application of a thin clear lacquer to prevent a marked change in the exposed surfaces which would increase the heat lost from them each hour. Table 72 gives an indication of the losses of heat from lacquered copper tubes.

205. The Installation of Piping. The success or failure of both steam and hot-water heating systems depends not only upon the selection of the proper sizes of pipes for the various portions of the system but also upon the care and skill used in the installation of the separate sections of pipe.

TABLE 72

HEAT LOSSES FROM BRIGHT COPPER TUBES HAVING ONE COAT OF THIN CLEAR LACQUER*

Btu per hour per linear foot per degree Fahrenheit temperature difference; Room Air 70 F

Nominal Pipe Size	Hot Water, Type-*K* Copper Tube			Steam, Standard Pipe Size Pipe		
	150 F	180 F	210 F	227.1 F	297.7 F	337.9 F
	Temperature Difference			Temperature Difference		
	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.265	0.282	0.307	0.401	0.461	0.478
¾	0.356	0.373	0.414	0.477	0.571	0.578
1	0.437	0.463	0.507	0.598	0.681	0.710
1¼	0.537	0.554	0.614	0.700	0.812	0.840
1½	0.612	0.645	0.714	0.833	0.966	0.990
2	0.762	0.818	0.892	1.005	1.164	1.201
2½	0.937	0.991	1.085	1.178	1.361	1.420
3	1.025	1.135	1.270	1.400	1.625	1.700
3½	1.250	1.318	1.442	1.580	1.845	1.905
4	1.400	1.480	1.556	1.750	2.040	2.130

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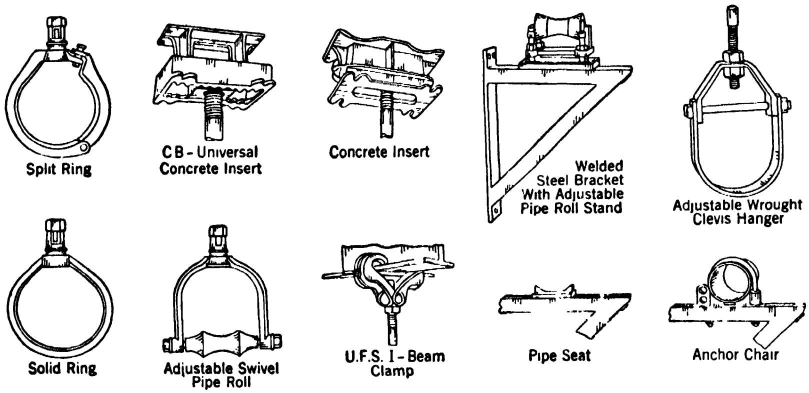
* Derived from Data published in *Heating, Piping and Air Conditioning*, paper by R. H. Heilman, 1933.

All internal burrs in pipe, caused by pipe-cutting tools, should be removed by reaming. Only pipe-threading dies which will cut perfect threads should be used. All joints must be made tight, a minimum amount of pipe-joint compound being applied to the threads of the pipes and nipples. The joint compound should not be applied to internal threads of fittings as it will be forced inside of the line to cause trouble in return lines and traps. Unless care is taken to remove any accumulations of dirt and other foreign materials from the pipes, trouble in the system is likely to occur. Traps of all sorts should not be put into operation until the greater portion of dirt and foreign materials have been carried out during the initial operation of the system.

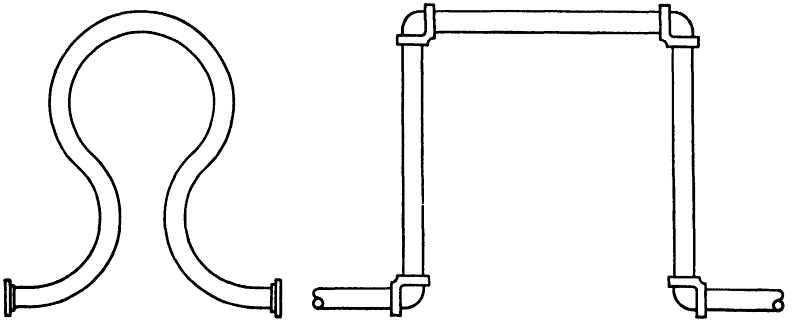
Pipe lines require adequate support so that they will not sag to produce undue stresses in them. Sagging also causes pockets in the lines so that they cannot be completely drained. Accumulations of water in pockets caused by sagging interfere with the flow of steam. Suitable types of pipe hangers and supports are shown by Fig. 150.

All pipes must be properly pitched, depending upon the conditions, in order that air and water may be removed. When water accumulations interfere with steam flow the water may be carried with the steam, provided that the velocity is great enough, to produce a disagreeable noise known as "water hammer." Water hammer under extreme conditions is dangerous as the slugs of water carried by the steam may cause a rupture of either the pipe or its fittings. Where counterflow of steam and water must occur within a pipe, the size of the pipe must be greater than that obtained by the methods of Art. 223, if water hammer is to be avoided. Pipe sizes are designated in Table 77, Chap. 10, for parts of steam heating system where counterflow of steam and water generally occur.


Split Ring
CB-Universal Concrete Insert
Concrete Insert
Welded Steel Bracket With Adjustable Pipe Roll Stand
Adjustable Wrought Clevis Hanger
Solid Ring
Adjustable Swivel Pipe Roll
U.F.S. I-Beam Clamp
Pipe Seat
Anchor Chair

(*Grinnell Co., Inc.*)

FIG. 150. Pipe hangers and supports.



Double Offset Expansion U Bend

U Bend with Four Fittings

FIG. 151. Expansion pipe bends.

206. Pipe-Line Expansion. Steam and hot-water pipes are erected with the materials at the temperature of the surrounding air. When such lines convey a heated medium they change in length as the result of the temperature rise of their materials. In high-pressure lines, the pressure of the material conveyed also affects the length, owing to the elastic properties of the materials. In lines operating with pressures less than 50 psig the effect of pressure other than its effect on temperature may be ignored. Provision must be made for change of length with temperature variations, as excessive stresses may be set up in the line and it may buckle to change its pitch to an improper one in places.

Provision may be made for pipe-line expansion (1) by the turning of certain fittings on their threads, (2) by the placement of bends in certain pipes, (3) by the use of expansion loops constructed of short pieces of pipe and elbows, and (4) by the use of any of the types of expansion joints.

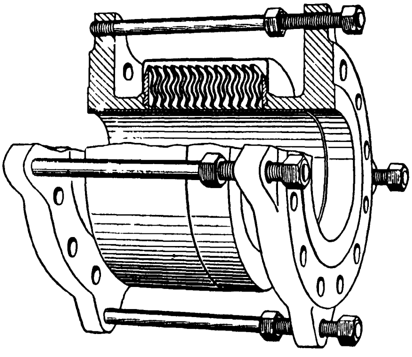


FIG. 152. Packless expansion joint.

Dependence is often placed on method 1 in low-pressure heating installations even though corrosion of pipe threads may prevent the necessary movement. Schemes 2 and 3, illustrated by Fig. 151, are self-explanatory. Figure 152 illustrates a packless type of expansion joint.

In low-pressure steam heating and in hot-water heating the change of pipe length may be based on the coefficients of Table 2, Chap. 1, without allowances for the effects of pressure within the pipe.

Example. A steel pipe line 100 ft long was erected when the temperature of the materials was 70 F. Find the amount of linear expansion when the line is filled with steam at an average temperature of 225 F.

Solution. The increase of length is

$$100 \times 12 \times 0.0000063 \times (225 - 70) = 1.17 \text{ in.}$$

207. Boiler-Piping Details. The sizing, the arrangement, and the fabrication of the outlet and the return connections are matters of great importance in the installation of both steam and hot-water boilers.

In general, all the boiler outlets provided by the manufacturer should be used, and pipe connection to each outlet should be of the size of the tapping. Restricted outlets cause undue losses in the pressure producing flow of the heating medium, and in steam boilers these may be a cause of entrainment of water with the steam discharged. All the return tappings provided should be used, and, in general, the pipe connections at the boiler should be the full size of the tappings pro-

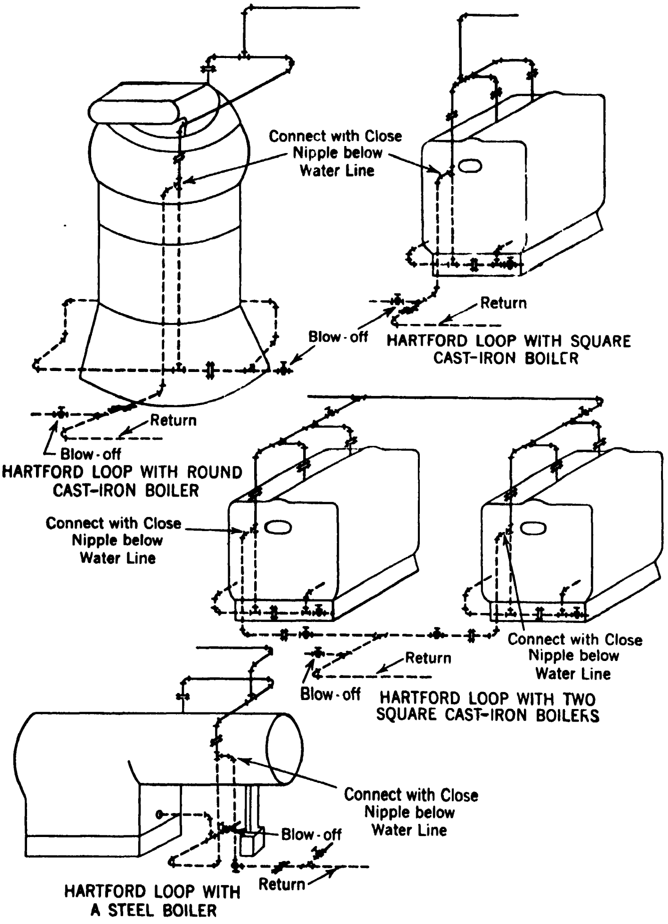

Connect with Close Nipple below Water Line
Return
Blow-off
HARTFORD LOOP WITH SQUARE CAST-IRON BOILER
Return
Blow-off
HARTFORD LOOP WITH ROUND CAST-IRON BOILER
Connect with Close Nipple below Water Line
Connect with Close Nipple below Water Line
Return
Blow-off
HARTFORD LOOP WITH TWO SQUARE CAST-IRON BOILERS
Connect with Close Nipple below Water Line
Blow-off
Return
HARTFORD LOOP WITH A STEEL BOILER

Fig. 153. Steam heating boilers with a Hartford loop in the return connections.

vided. Boiler-return connections for gravity-flow hot-water systems should be the same size as those of the water-heater outlets. Steam boilers may have their return-pipe connections smaller than those used at their outlets. However, as these return connections are at the low point of the water space in a boiler it is desirable to make all the return piping around the boiler large enough to take care of some accumulations of dirt, sand, and scale which may be washed down from the interior of the boiler. These larger sizes of pipe reduce the likelihood of failure of water to return to the boiler from the system because of clogging of the return connections.

Heating boilers are immobile objects which require that provision be made at their outlets for expansion of the piping; otherwise excessive stresses may be developed at the outlet tappings. This is accomplished in small- and medium-sized boilers by means of a single-swivel arm from each outlet attached to a common header as in Fig. 153. For very large boilers a double-swivel arrangement may be necessary.

Symbols suggested for the preparation of heating and air-conditioning drawings are given in Table 134 of the Appendix and will be of aid in connection with an understanding of Fig. 153. With steam boilers it is desirable that the outlets all be connected to a common header as is shown by Fig. 153. The header should be made of pipe sufficiently large to act as a steam separator. A drip or drain connection is necessary between the header and the return connections to the boiler. The function of the drip is to return speedily slugs of water thrown out with the steam to the boiler without having the water carried through the supply and return mains. Flanges should be installed in the pipes just above each outlet and in the return connections near the boiler so that the boiler may be removed for either repairs or replacement with the minimum amount of disturbance to the piping.

208. The Hartford or Underwriter's Loop. All the boilers shown by Fig. 153 have the protective device in the return piping known as the Hartford or Underwriter's Loop. The loop is used only with steam boilers as it is neither needed nor desirable in connection with hot-water boilers. The loop is constructed of pipe and fittings and costs only the price of the materials and the labor of installation.

A loss of steam pressure occurs in the supply mains as the steam flows away from the boiler. In closed-circuit systems the end of the steam main is connected through the return main to the boilers. Owing to the loss of steam pressure in the main the boiler pressure will cause the water level to stand higher in all drops to the return connections than the normal boiler water level. Under bad conditions of operation a considerable amount of water may be forced out of the boiler through the return connections and thus lower the water level in the boiler to a point where damage to the boiler may occur. The upper portion of the loop is attached by means of a close nipple to the equalizer connection between the boiler header and the return yoke. The nipple connection is placed so that the water level in the boiler cannot be lowered more than from 2 to 4 in. below the normal water line. When the top part of the close nipple is uncovered by water, steam from the header exerts its pressure on the return piping and further loss of water from the boiler is prevented.

The Hartford or Underwriter's Loop cannot prevent water from being lost from a boiler by evaporation; its sole function is to prevent loss of boiler water by backward flow into the return mains. Recommended sizes of equalizer pipes between the header and the return connection are given in Table 73.

TABLE 73

SIZES OF HARTFORD LOOP EQUALIZERS

Boiler Grate Area, Sq Ft	Pipe Size, In.
4 or less	1½
4 to 15	2½
15 and more	4

PROBLEMS

1. Find the heat losses per hour from 800 lin ft of 6-in. commercial steel pipe when bare and when insulated with 1.08-in.-thick 85 per cent magnesia material. The pipe is filled with saturated steam at 20 psia, and the surrounding air temperatures are 80 F with bare pipe and 65 F with covered pipe.

2. Find the efficiency of one-inch Air-Cell pipe covering when applied to 20 lin ft of 4-in. steel pipe filled with steam at a temperature of 250 F when the surrounding air is at a temperature of 85 F.

3. Calculate the heat losses from 1200 lin ft of 2-in. bright copper tubing when operated with water at an average temperature of 180 F when the surrounding air is at 70 F.

4. Compute the linear expansion of 125 ft of steel pipe when it conveys steam at 350 F and also when the steam temperature is 240 F. The pipe installation was made when its temperature was 75 F.

CHAPTER 10

HEATING WITH STEAM

209. Classification of Steam Heating Systems. Common present-day types of steam heating systems are one-pipe air-vent, one-pipe vapor, two-pipe vapor, and vacuum arrangements. One-pipe installations have, at the bottom of each heat-dispersal unit, a single-pipe connection through which the supply of heating steam and the removal of the resulting condensate is made. Two-pipe systems have separate inlet and outlet connections to each heat disseminator, and the problem of the counterflow of steam and condensate is not so involved as it is in one-pipe systems.

In gravity systems the flow of condensate through the return piping into the boiler is produced as indicated by the name, and no pump of any sort is required to handle it. Mechanical systems employ either a condensation pump or a vacuum pump to force the returned condensate either into a boiler or a feedwater heater. In all mechanical-system arrangements the condensate flows to a receiver from which it is removed by a pump. The water line of the receiver must be low enough so that adequate drainage from the system is obtained. The height of the boiler water line is immaterial except as it affects the head against which the pump discharges the water. When gravity return of condensate is made to a boiler all radiators and the ends of all supply mains must be properly located above the level of the water in the boiler. Mechanical systems permit the location of heating units below the level of the boiler water line and also the use of boilers operating with steam pressures greater than those desired in the system. Vacuum systems are of necessity two-pipe mechanical layouts. All others enumerated may be operated with either gravity or mechanical return of the condensate. Any of the systems listed may be installed with basement mains allowing the steam to feed upward through risers to the radiators of the building or with attic steam mains which supply steam to risers carrying downward through the structure.

The term "relief" applied to either a one-pipe or a two-pipe system indicates that special provisions are made for draining supply piping at various points into the return main in order to relieve the supply piping of condensate. Such a procedure allows the use of smaller sizes of the various sections of the steam supply piping than would otherwise be required. Air-line set-ups are modifications of air-vent systems whereby special air valves together with an exhauster and connecting air piping are used to free radiators of air more positively and quickly than when ordinary air valves are used. Such an arrangement eliminates the discharge of odor-bearing air and water from radiators into occupied spaces.

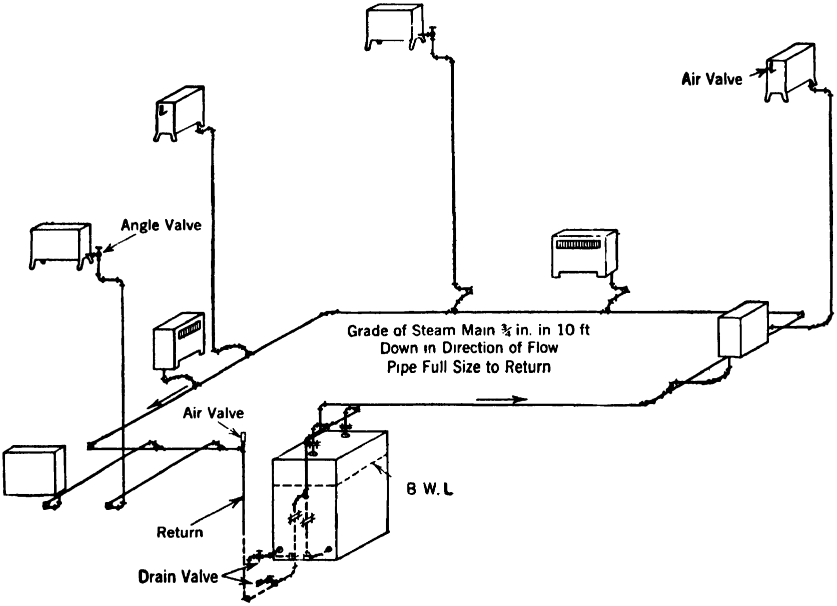


Fig. 154. One-pipe air-vent steam-heating system.

210. One-Pipe Air-Vent Systems. These systems having only a single-pipe connection to a radiator require air vents for the supply mains and each radiator. The vents are automatic valves which discharge the air which accumulates in the system from air inleakage at different places or as a result of the steam carrying air and other noncondensable gases. Unless the radiators and supply mains are kept free from air they will not be completely filled with steam, and the system will not function properly. Typical sketches of one-pipe steam systems with gravity flow are shown by Figs. 154 and 155. The first illustration is of an up-feed system; the second, of an overhead down-feed installation. These systems, when properly laid out and installed, should not require a boiler steam pressure of more than 2 psig.

In the up-feed arrangement the supply main is taken from the boiler header and rises close to the basement ceiling. From this point the main is pitched downward in the direction of steam flow with a grade of not less than $\frac{1}{4}$ in. in 10 ft. The main may be carried in a single loop about the basement, or it may be divided into two or more parts as the building arrangement may require. The steam and condensate in the main flow in the same direction and the divisions of the main should not be reduced in size, as branches to risers are taken off unless drip connections to the return piping are provided at each point where the pipe size is reduced. The simplest method of installing one-pipe mains is to keep the divisions of the main uniform at the sizes necessary for the total radiation served by any part of them. All ends of the main should be maintained at least 18 in. above the normal boiler water line in order to allow for the head of water in the drops at those points. It often becomes necessary to place the boiler in a pit in order to lower the boiler water line with respect to the ends of the main. When the main is long or the basement head room is limited, relaying the main, Fig. 156, helps to solve the problem. A drain connection at the bottom of the rise is always required; otherwise the pocket will fill with water. The steam main and its divisions should be designed for losses of steam

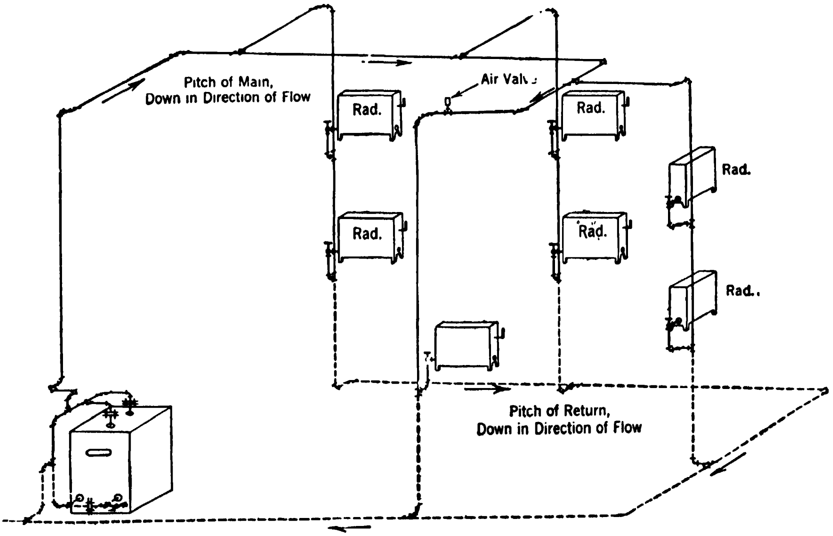


FIG. 155. Overhead downfeed one-pipe steam heating system.

pressure not to exceed 1 oz per 100 equivalent ft of length. For most installations the minimum size of a steam-supply main should not be less than 2-in. pipe.

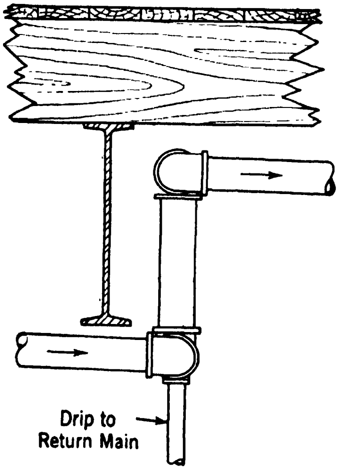


FIG. 156. Steam main relay with drip connection.

Return mains are designated as either wet or dry. A wet-return main is usually but not always below the boiler water level and is filled with water returning to the boiler. A dry-return main in a steam-heating system carries both condensate and air and sometimes may be filled with steam although this is not a desirable condition. A dry-return main may be either above or below the boiler water level. With gravity return of the condensate to the boiler the dry-return main must be above the level of the water in the boiler. A return line above the boiler water level can be made to function as a wet-return main by the scheme of Fig. 157.

All branch connections from the main to the riser pipes are to be taken from either the top or side of the main with a uniform pitch to facilitate the flow of the condensate and also to avoid any water pockets. In up-feed systems without riser reliefs the branches pitch upward from the steam main toward the risers at least $\frac{1}{2}$ in. per ft. Also the runouts from the risers to the radiators are given the same pitch upward toward the radiator valve. Counterflow of steam and

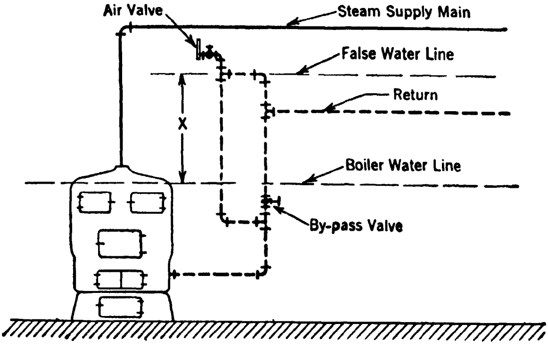


FIG. 157. False water line.

water occurs in the branches from the mains, the risers, the runouts, and the radiator valves of up-feed systems. The sizes of these parts must be such that steam and water flow will not interfere. With an overhead steam main each branch from the main pitches downward toward its riser and the branch connection from the bottom of each riser to the return main slopes in the direction in which the condensate flows. Counterflow of steam and water occurs in radiator runouts of the down-feed one-pipe arrangement.

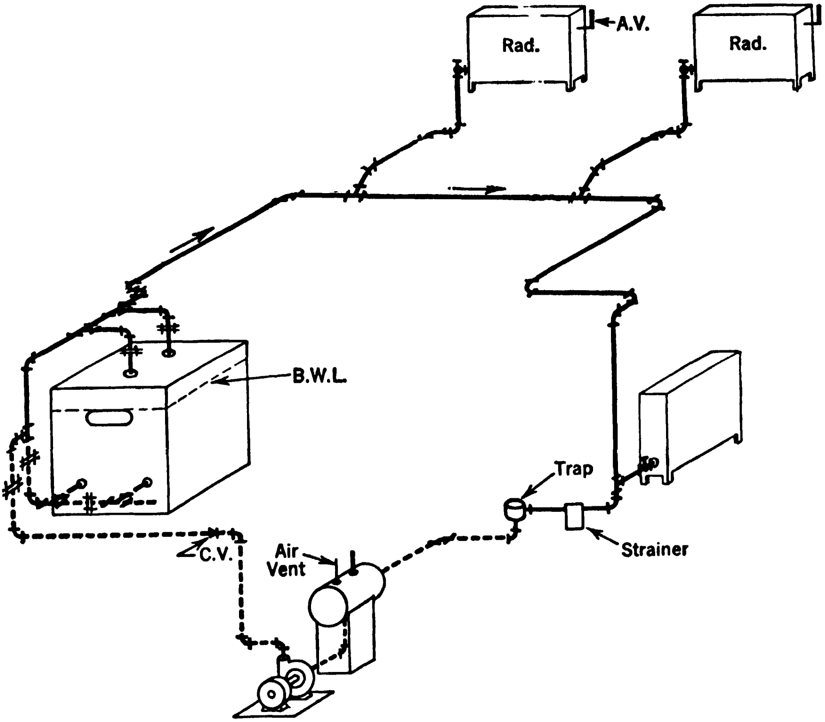


Fig. 158. One-pipe air-vent mechanical steam heating system.

One-pipe air-vent systems are the simplest and cheapest to install of all the types in use. When properly laid out and installed they give satisfactory results. The overhead down-feed one-pipe systems find usage in tall buildings. As the steam and water flow in the mains and the risers in the same direction, smaller pipes are permissible in the down-feed than in the up-feed arrangement.

211. Systems with a Condensation Pump. Either of the gravity-return steam systems just described may be converted to a mechanical system by the addition of a condensation pump. Figure 158 is an example of a one-pipe steam system so arranged. A pump is necessary when radiators must be placed below the boiler water line or at an insufficient distance above it. Building limitations often prohibit a boiler pit so that a pump becomes necessary. A pump is also required when the boiler steam pressure is greater than that of the supply mains.

Condensation pumps have a receiver in which the condensate brought back by the return lines accumulates. When a sufficient quantity of water has collected the pump is started by the action of a float within the receiver or by the tilting action of the receiver. Figure 158 gives an idea of the appearance of a condensation pump of the former type.

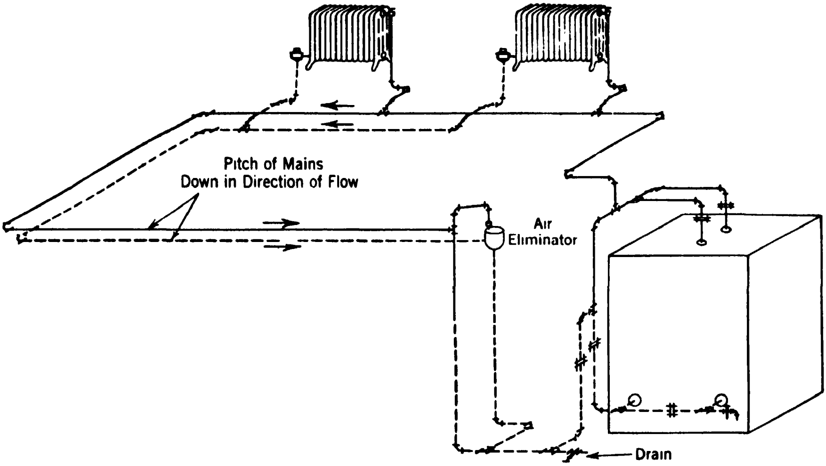


FIG. 159. Elementary vapor steam heating system.

212. Vapor Systems. The designation vapor system is applied to layouts, Fig. 159, which function as follows:

1. The boiler steam pressure ranges from a few ounces above atmospheric as a maximum to pressures less than atmospheric. The subatmospheric pressures are obtained by condensation of steam in the radiators when the boiler furnace fire is declining in activity. The extremely low steam pressures employed are the principal distinguishing feature.

2. The condensation is generally returned to the boiler by gravity. With certain vapor systems a pump may feed the water to the boiler, but the pump does nothing to create a vacuum in the system.

3. All the air vented from the radiators and the piping is discharged from the return mains except in one system.

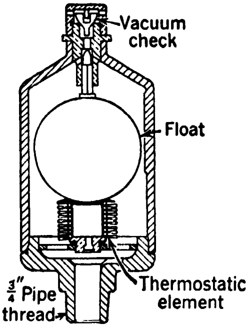
Most vapor systems are two-pipe installations. However, there is a one-pipe vapor system which operates with a few ounces of steam pressure. The features are those of a one-pipe air-vent system with the following exceptions. The radiator valve is a plug cock arranged to give the minimum amount of obstruction to water and steam flow when open. The air valves on the radiators are special units designed to give quick relief of air; they are of the non-return type to prevent expelled air from re-entering the system.

Two-pipe vapor systems may be divided into two general groups, non-return and open or atmospheric systems. The non-return systems have air eliminators at the ends of the dry returns near the boiler which, after the system has been freed of air, prevent its immediate return. When free of air the system may operate with steam pressures less than atmospheric for periods of time which are dependent upon the tightness of the piping connections, etc., and the heating load existent.

Open systems have the return piping under atmospheric pressure at all times as the air vents are not provided with check valves to prevent the inflow of air. The pressure of the steam in the supply mains is kept slightly above atmospheric. Open systems may return the condensate to the boiler either by gravity or by a pump, or they may allow it to waste at a drain, as is done in some district-heating installations.

213. Features of Vapor Systems. The heat-dispersal units employed by vapor systems include tubular and wall radiators, pipe coils, finned sections, and various forms of convectors. Some form of graduated-control valve is placed at the inlet tappings of radiators and pipe coils in order that the steam supply to the unit may be adjusted to give the desired heat output. The valve should be of the packless type in order to eliminate air inleakage around the valve stem when the pressure within the radiator is less than that of the atmosphere. The steam flow in the radiator is through the top row of nipples and downward through the radiator sections. The radiator may be only partially filled with steam except when the demand for heat is maximum, and then it may be filled with steam. When the radiator is not completely filled with steam portions of the surface not in contact with steam are warmed by condensate flowing down the inside. This allows heat to be abstracted from the condensate, which is of importance when the water is not returned to the boiler. The outlet connection to the return piping may be made at either end of the radiator and is always at a bottom tapping. Free discharge of the air and the condensate should be obtainable through the outlet connections of all radiators. Generally steam is prevented from flowing through into the return piping either by a trap which will pass both water and air or by a water seal which is arranged to allow the escape of air and condensate. A few systems have neither radiator traps nor seals; they have about 20 per cent more surface in the radiators than is ordinarily required, and the graduated control inlet valve is adjusted to prevent the radiator from being completely filled with steam under any conditions.

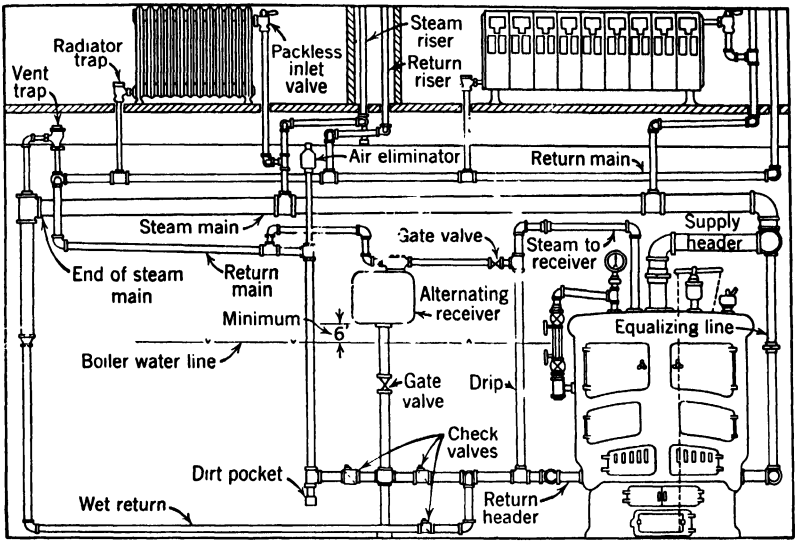
214. Vapor-System Installations. Vapor systems are installed as both up-feed and down-feed plants. A dry-return main is necessary when the condensate is returned by gravity to the boiler, as the air from the radiators and supply piping is discharged into the dry-return mains from which it is finally vented. Wet-return mains are preferable for carrying the condensate back to the boiler, although a dry-return main may be used for this purpose. In up-feed systems the end of the steam main may be brought back near the boiler where it is dripped into a wet-return. All down-feed steam-supply risers should be dripped through either a trap or a seal into a return main. In the up-feed arrangement the supply main is taken to a high point above the boiler from which it grades downward in the direction of steam flow. The branches from an up-feed main are pitched upward toward the risers served unless the riser is provided with a properly sealed drip connection at its bottom; the branch then pitches downward toward the riser. Counterflow of steam and the riser condensation occurs in all up-feed risers and also in the branches from the main to up-feed risers when the supply risers are not dripped at the lower end. The bottoms of all return risers are connected to the return main through branches which are pitched downward from the risers to the return main. A dry return should pitch downward in the direction of flow toward the boiler. The air vent in the dry-return main is located at the point where the dry return drops below the boiler water line. The simplest air vent for a dry return is a $\frac{3}{4}$-in. check valve arranged to open outward. Most systems which employ patented specialties have an air eliminator, Fig. 160, which allows the discharge of air but prevents its return into the piping. Wet-return mains are pitched downward toward the boiler from their point farthest away from it, Steam mains should have a minimum pitch of $\frac{1}{4}$ in. in 10 ft, branch and runout connections $\frac{1}{2}$ in. per ft, dry-return mains 1 in. per 20 ft, and wet-return mains 1 in. per 40 ft.



(*Sarco Co., Inc.*)

FIG. 160. Air eliminator.

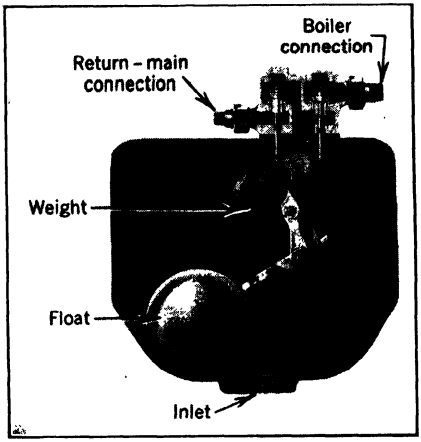
All supply piping must be large enough to limit the steam-pressure losses to 1 oz or less per 100 ft.



(*Sarco Co., Inc.*)

FIG. 161. Vapor heating system with alternating receiver.

215. Alternating Receivers or Return Traps. In order to secure positive return of the condensate to the boiler at all times a device known as an alternating receiver or return trap is often used in vapor systems. Figure 161 gives the details of a closed vapor system which does not permit the return of air into the dry-return main and which has an alternating receiver.



(*Sarco Co., Inc.*)

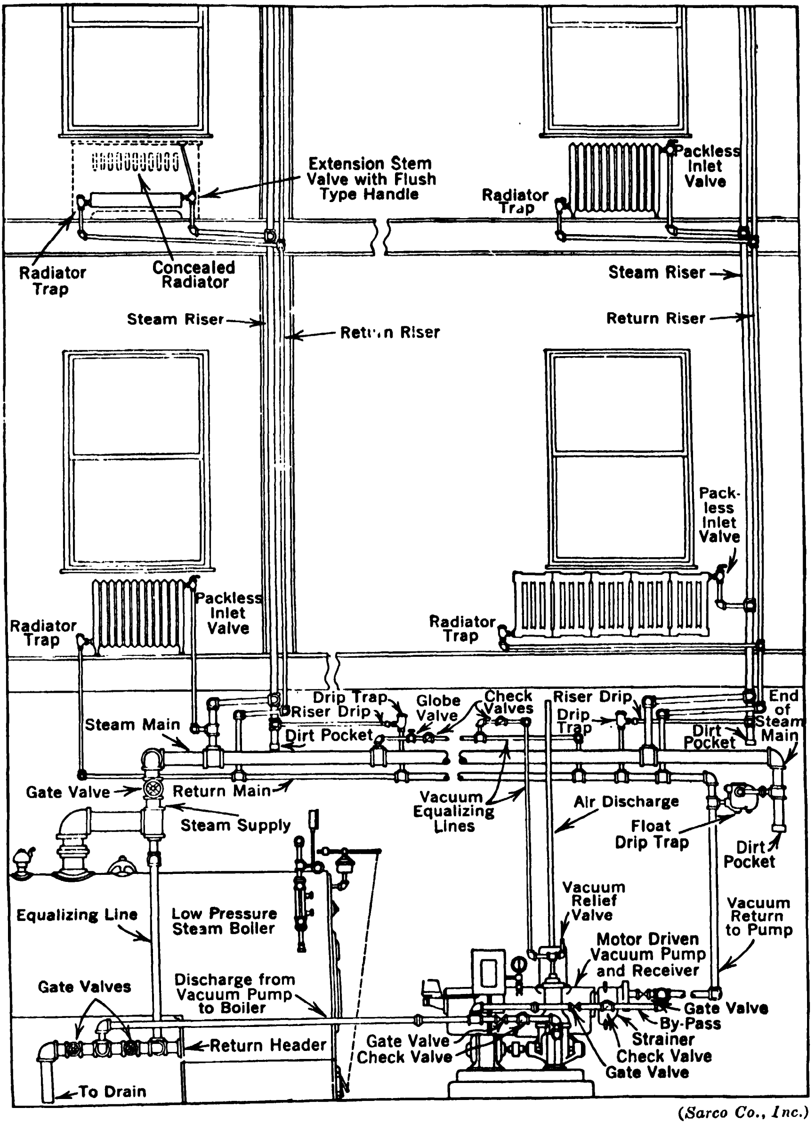
FIG. 162. Alternating receiver.

Figure 162 gives a sectional view of a return trap or receiver. The unit has a float chamber in which a hollow float rises as water accumulates in it. The float is attached to a weight and lever mechanism connected to a balanced steam valve which is in a pipe leading from the boiler steam outlet. The float when lifted brings the mechanism to a point where it trips. The steam valve is quickly opened and at the same time the trap top vent to the return main and the air eliminator is closed. The action just described equalizes the pressure in the trap with that of the boiler, and the collected water in the trap flows to the boiler by gravity. When the trap has been emptied of water the float causes the mechanism to open the air vent and to close the steam valve so that more water may accumulate. Traps of this kind are installed with the bottom of the trap from 6 to 18 in. above the normal boiler water line. The elevation of the trap is dependent upon the particular make of equipment used.

216. Vacuum Systems. These systems are distinguished from certain other two-pipe steam systems in that a pressure less than atmospheric in the return piping is mechanically produced by means of a vacuum pump, Fig. 163. Each radiator of the system is fitted, at its return outlet, with a thermostatic trap which will allow both air and water to pass but which closes against the flow of steam. All drips into return mains, from mains and supply risers, must pass through traps which prevent the loss of steam but pass both air and water.

Direct cast-iron tubular and wall radiators, pipe coils, and convectors of various kinds are suitable in vacuum heating. The radiator inlet valves should be of a packless type with graduated control although other valves are possible. The details of the supply and return piping are similar to those of some vapor systems, and the systems may be installed with either up-feed or down-feed supply mains.

For the best results the vacuum pump should be located so that the return piping grades uniformly downward toward the receiver of the unit. This allows gravity flow of the condensate to the pump. Sometimes lift fittings are permissible in the return piping when it is necessary to raise the return piping from one level to another, but lift fittings are to be avoided whenever possible. The function of the vacuum pump is to withdraw air which is vented from the supply piping and radiators into the return mains. The vacuum pump withdraws the condensate from the receiver and discharges it into low-pressure heating boilers or into the feedwater heater when high-pressure boilers supply steam to the heating system through a pressure-reducing valve. The air withdrawn from the system by the pump may be separated from the condensate at the vacuum pump and discharged to the atmosphere or it may be discharged with the condensate and expelled from the storage reservoir into which the pump discharges. This receptacle may be either the feedwater heater or an overhead tank from which the water may flow by gravity to a low-



(*Sarco Co., Inc.*)

FIG. 163. Vacuum heating system.

pressure boiler. In any event, the air and condensate must be separated and the collected gases must be discharged before the condensate is fed into the boiler supplying the steam.

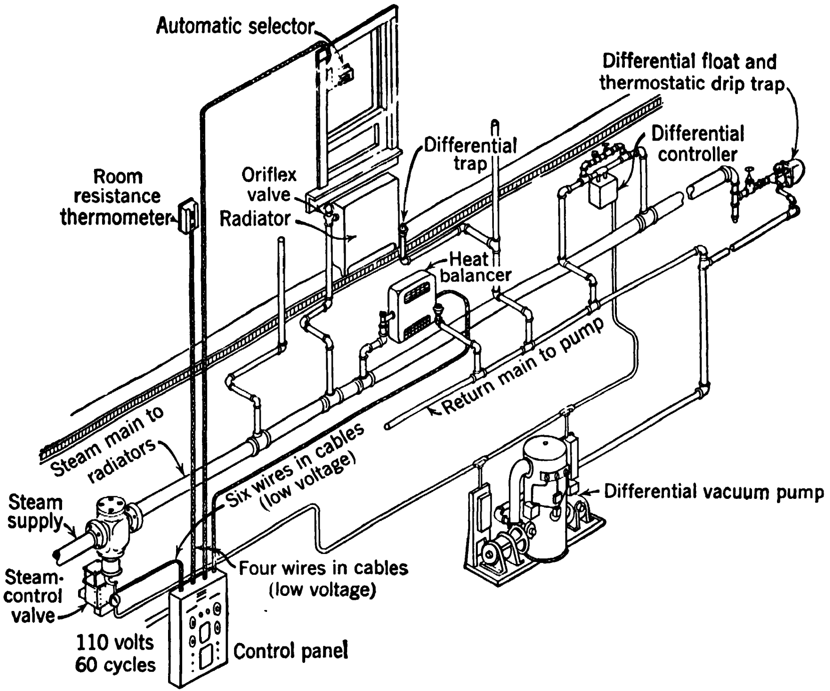
Vacuum systems are adapted to large buildings and also those covering considerable ground area. The action of the vacuum pump is to produce free circulation of steam by the maintenance of a definite pressure differential between the supply and the return piping. This action enables the more rapid venting of air to take place, as steam pressure alone is not necessary to force the air out. Positive discharge of the condensate is also effected, and radiators may be located below the level of the boiler water line.

Vacuum steam heating systems may be separated into two general groups: (1) return-line systems and (2) differential or sub-atmospheric systems.

217. Vacuum Return-Line Systems. Return-line installations generally have a vacuum maintained in the return piping without any attempt to produce pressures less than atmospheric in the radiators and the supply piping. Figure 163 illustrates a typical up-feed system with water-type radiators, graduated-control valves at the radiator inlets, thermostatic traps at the radiator outlets, and suitable drip traps to pass water and air at the ends of the steam mains.

The steam mains should be pitched in the direction of flow not less than 1 in. in 20 ft, and the dry-return mains should have a pitch of at least 1 in. in 40 ft. Steam mains and other pipes should be designed with the maximum allowable pressure loss not to exceed 1 oz per 100 ft of pipe. The traps at the outlets of radiators do not function exactly the same as those of vapor systems; consequently their capacities for a given size are somewhat different. The pressure differential between the two sides of a radiator trap must be at least equivalent to 8 oz. The vacuum pump should maintain a vacuum of at least 10 in. of mercury in its receiver. This vacuum may be reduced to 2 in. of mercury at the radiator trap farthest removed from the pump. The displacement of the vacuum pump must be sufficient to handle not only the condensate and air discharged into the return piping but also the vapor produced by the re-evaporation of condensate passed by the traps. All radiator and drip traps should operate to prevent steam entering the return piping; otherwise the maintenance of a vacuum by the pump becomes increasingly difficult as the leakage of steam increases.

218. Differential-Vacuum Heating Systems. These systems are operated to vary the heat output from the radiators in accordance with the demands of the heating load. The temperature of the room air is regulated by changing both the temperature and the quantity of the steam supplied to the radiators. The steam is supplied continuously but with varying rates of flow and temperature, which is controlled by adjusting its pressure in the supply piping. Dependent upon weather conditions the steam in heat-dispersal units may exist with temperatures and specific volumes corresponding to pressure conditions ranging from 2 psig to 25 in. of mercury vacuum.



(*C. A. Dunham Co.*)

FIG. 164. Control equipment of Vari-Vac differential heating system.

The details of sub-atmospheric heating plants differ from return-line systems principally in the matter of control equipment. There is also a slight difference in the details of radiator inlet valves and traps. Each steam-supply valve at the heat-dispersal units has either a fixed or an externally adjustable orifice so that the steam distribution between the various units may be properly adjusted. The radiator traps have a pressure range of from 15 psig to 25 in. mercury, which is greater than that necessary in a return-line vacuum system.

Important parts of the control equipment, Fig. 164, are: (1) the steam-control valve, (2) the control panel, (3) the heat balancer,

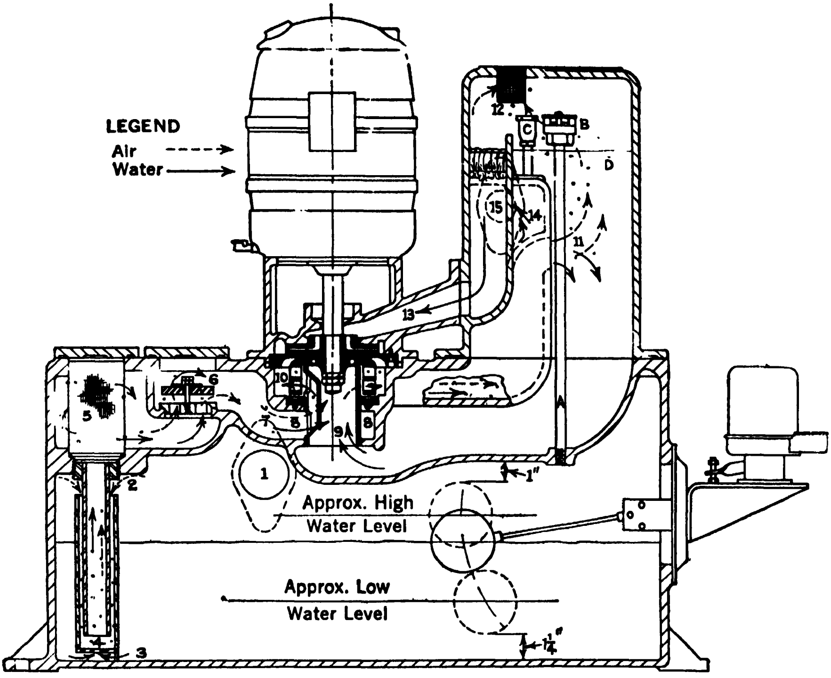
(4) the selector, (5) either a room thermostat or a room resistance thermometer, (6) the differential controller, and (7) the vacuum pump.

The steam-control valve operates to vary the flow of steam, with changes of load, to the supply piping and functions under the action of the control panel which transmits the combined actions of the heat balancer, the selector, and the room thermometer. The heat balancer is a form of convector having a pair of resistance thermometers one of which is located above its heating element and one below it. The combination of the two thermometers measures the air-temperature differential existing on the two sides of the heating element, and the heat balancer thereby gives an indication of the heat supply to the system. The heating demand is measured by the selector, attached to the inside surface of a window, which is influenced by the effects of outside-air temperature and wind velocity. In connection with the control panel the room thermometer functions to compensate the effects of the selector and the heat balancer. The differential controller connected to the supply and the return piping is actuated by the pressure differential existing between the two lines and functions to start the pump when the difference is small and to stop the pump when conditions are satisfactory. Air and non-condensable gases are eliminated from the piping and the radiators by action of the vacuum pump, which also discharges the condensate from the system to the heating boiler or any storage facility.

Commercial heating boilers, pipe, and pipe fittings are used in the construction of sub-atmospheric systems. The sizing of pipe is done in the same manner as for other vacuum systems, and the pitch of the supply and return mains is downward in the direction of flow in the amount of $\frac{1}{2}$ in. per 10 ft. Branch connections between mains and risers should be pitched $\frac{1}{2}$ in. per ft to give adequate drainage where counterflow of steam and water takes place. Lift fittings should not be used in the return mains, the only exception being when such an arrangement is placed near the vacuum pump.

219. Vacuum Pumps. Air may be removed, and condensate from the returns of vacuum systems may be handled by various types of vacuum pumps. Vacuum pumps may be high-pressure steam-driven reciprocating units, motor-driven rotary units, or rotary units which embody both a motor drive and a low-pressure steam-turbine drive. In the last type the drive alternates according to operating conditions.

Figure 165 illustrates a motor-driven unit. The air and condensate from the heating system enter the receiver at 1. The air leaves the receiver at 2 and is mixed with water, passing from the receiver at 3, in the mixing manifold at 4. Both the air and water pass through the strainer 5 and the vertical check valve 6. The air and water flow through passage 7 to two ports at 8. At 9 prime water from the pump casing enters two openings which alternate with the air and water openings shown at 8. The housing which contains ports 8 and 9 is stationary. The pump impeller, shown as 10, revolves around the stationary port housing and picks up slugs of air and water at ports 8 and prime water at ports 9. The mixed air and water entering at ports 8 and the prime water entering at ports 9 are discharged by the impeller through the discharge channel and are separated at 11. The separated air rises to the vent at 12. The water flows over a baffle plate in passage 13 to the separate boiler feed-pump impeller. The water discharge from this impeller is through channel 14 to the discharge outlet at 15 where the water leaves the unit and flows to the heating boiler through the connecting piping.



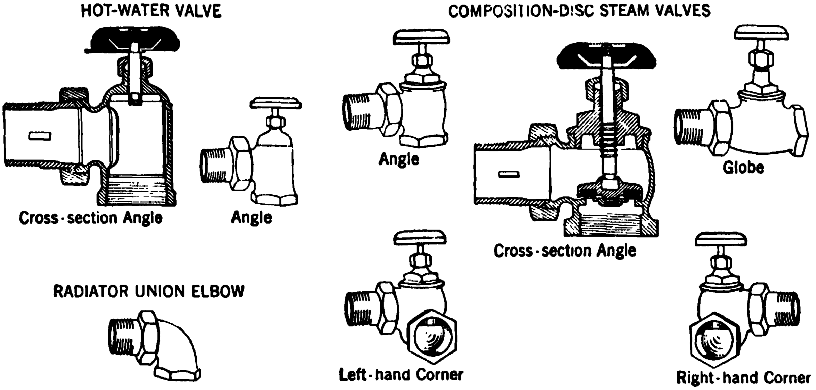
(*Chicago Pump Co.*)

FIG. 165. Return-line vacuum pump.

A is an air-relief valve from the receiver, *B* is a check valve in the air-relief line, *C* is a vertical check valve to prevent water from leaving the pump casing when a vacuum exists in the boiler, and *D* is the level of prime water maintained at all times.

Piston vacuum pumps, either steam- or motor-driven units, should have a volumetric displacement at least 6 times that of the volume of condensate handled. Rotary pumps should be able to handle not less than 0.75 lb of condensate per hr per sq ft of attached equivalent radiation and 0.6 cu ft of air per min per sq ft of radiation.

Radiator traps should prevent steam from entering the return mains, and the piping should be tight if the desirable vacuum of 10 to 20 in. of mercury is to be obtained in return-line systems and up to 25 in. of mercury in differential-vacuum systems. A water spray at the vacuum-pump suction to condense steam which has leaked into the return lines is not desirable. Condensation of such steam is often required in order that a desired vacuum may be produced but it is accompanied by a loss of heat and excess water.

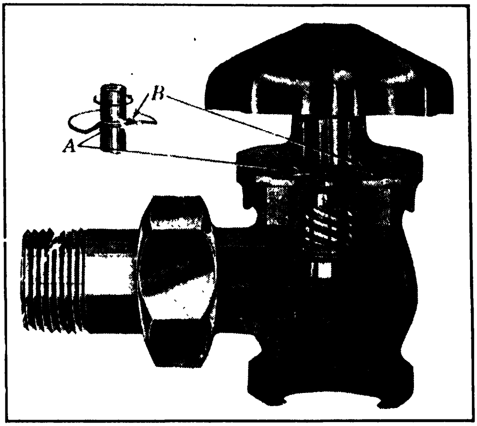


(*Crane Co.*)

FIG. 166. Radiator valves.

220. Radiator Valves. The control valves used with the radiators of any heating system are important items. The material of construction of valve bodies is usually brass, and they are made in angle, straight-away, and right- and left-hand corner patterns for both steam and hot-water service. The valve action in both opening and closing may be quick or slow. Valves are also made to give graduated or fractional opening with the amount of port opening indicated by a pointer attached to the valve stem or handle. Graduated-control valves cannot be used with one-pipe steam systems and are not generally used for hot water.

Figure 166 depicts the sections of angle radiator valves and indicates the external appearance of straight-away and corner steam radiator valves which have soft packing around the valve stem within the bonnet. The same valve may be used for hot-water radiators provided a small hole is drilled in the seat web to allow some water to circulate when the valve is closed, thereby preventing the water in the radiator from possible freezing. A packless-type angle radiator valve is illustrated by Fig. 167. These valves are also made with graduated control. Part *A* is a collar machined on the valve stem against which stainless-steel wafers *B* press to form a tight annular metal-to-metal seal against the leakage of steam, water, or air.



(*J. P. Marsh Co.*)

FIG. 167. Packless radiator valve.

All radiator valves should be provided with a union between the valve body and the nipple which is screwed into the tapping of the radiator tapping bushing. This arrangement facilitates the connection of the radiator to the supply piping.

221. Radiator Traps. These devices function to allow the escape of condensate and air from radiators and are used in connection with vapor, vacuum, and other two-pipe steam-heating systems. Radiator traps should be positive in their action of freeing the radiator of water and air but should always prevent the escape of steam. Present-day radiator traps are generally thermostatic devices built in various commercial forms, three of which are illustrated in Figs. 168 and 169. Thermostatic traps are comprised of a body, having an inlet opening and an outlet port with a seat, together with a cover to which may be attached a thermostatic element. The corrugated element is made of some alloy, such as bronze, and has sealed within it a volatile fluid

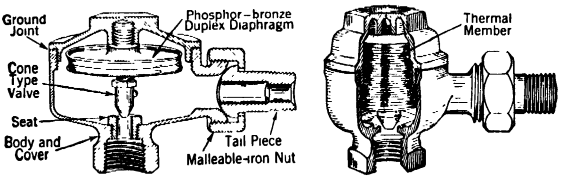
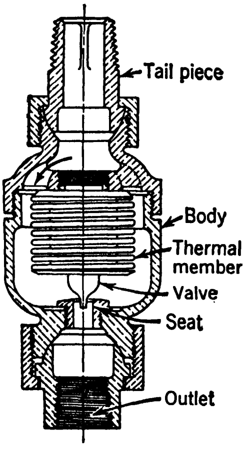


FIG. 168. Thermostatic radiator traps.

which is properly compounded to provide for the expansion and contraction of the element over a range of temperatures. When the thermostatic element is cool it is contracted and the outlet port is open.

Steam entering a cool radiator by reason of its pressure forces the air of the radiator through the open port of the trap to the return piping. Steam condensation also occurs, and the water thus produced is discharged by the trap until the radiator is filled with steam and the thermostatic element is heated. Heat from the steam then vaporizes enough of the volatile fluid to create a pressure within the unit sufficient to overcome the external pressure and the spring action of the corrugated metal which holds the closing member away from the outlet port. As soon as the element has a sufficient pressure within it, the port is closed and the escape of steam prevented. When the radiator is in normal operation, the thermostatic element does not alternately completely close and then fully open the outlet port but adjusts itself to operating conditions to permit an uninterrupted discharge of air and water from the radiator at a rate sufficient to keep the interior surfaces free from both.



(*Sarco Co., Inc.*)

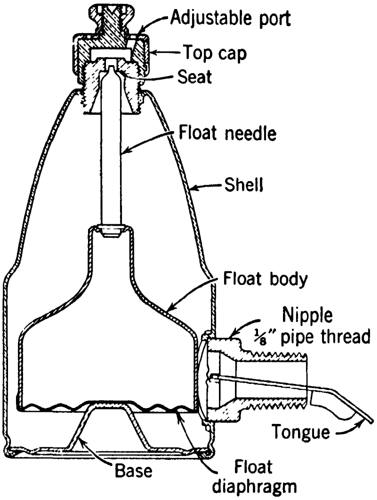
FIG. 169. Vertical trap for concealed radiation.

Thermostatic traps are made in angle, straight-away, and corner patterns for both high- and low-pressure duty. High-duty traps operate with pressures ranging from 15 psig to 25 in. of mercury vacuum. Low-duty traps function with pressures ranging from 15 psig to 15 in. of mercury vacuum.

Thermostatic blast traps are similar to radiator traps, but they have larger capacities and are suitable for higher operating pressures. Thermostatic blast traps may be obtained in angle, straight-away, and corner patterns. The trap illustrated in Fig. 144*a* is not suitable for use where air must be discharged.

222. Automatic Air Valves. One-pipe steam radiators require a vent to discharge the accumulation of air and other non-condensable gases within the radiator. Such devices are of various forms, but in general all such equipment embodies some form of a float and thermostatic arrangement.

The illustration of Fig. 170 shows a valve which has a body enclosing a float containing a volatile fluid. Air and other non-condensable gases pass through the opening in the seat and escape through an adjustable port in the top cap. The cap has six various-sized orifices, and the venting rates of the radiator air valves can thereby be regulated so that the different parts of the system will heat uniformly. The valve is attached to the radiator by a nipple screwed into a $\frac{1}{8}$-in. pipe tapping. As air is heavier than steam the proper location for the air valve would be at the bottom of the radiator in the end farthest from the inlet valve. Because of the danger of the air valve being flooded with water the valve is usually attached to the radiator at a point located at about two-thirds of its height. The hinged tongue entering the radiator through the valve tapping aids in freeing the valve of water by causing the formation of a drop of water between it and the radiator tapping boss to start drainage.

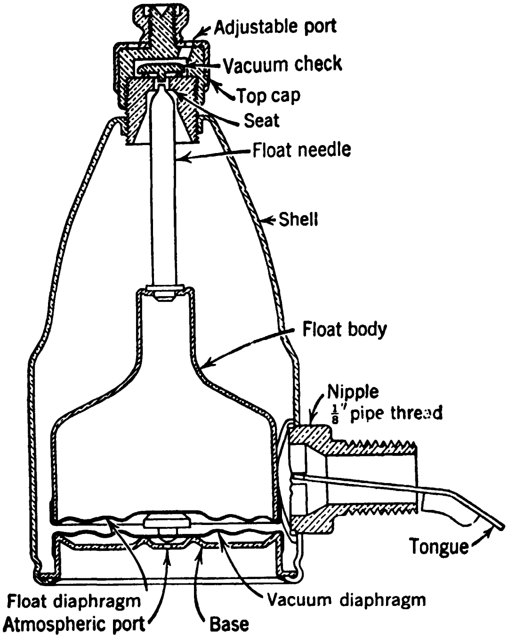


(*Hoffman Specialty Co.*)

FIG. 170. Radiator air valve.

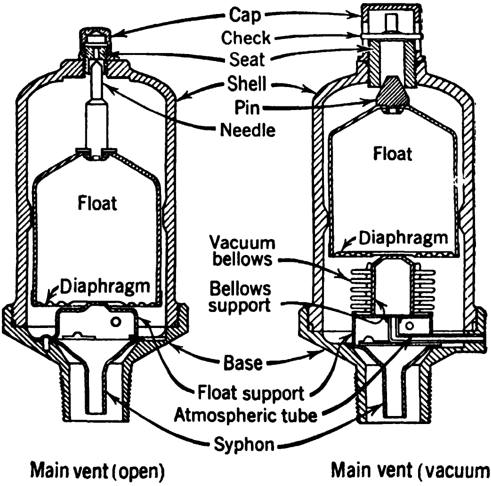
The bottom of the float contains a flexible diaphragm which is pushed downward when steam causes the volatile liquid within the float to vaporize. This action pushes the float upward to cause the valve pin to close the vent port when steam enters the shell. When the shell fills with water the float also lifts, thereby closing the vent to prevent the escape of water. When steam or water is not causing the vent to be closed, free discharge of air is secured.

Valves of this sort are made in the non-return or vacuum type as illustrated by Fig. 171. The construction illustrated in Fig. 171 includes the features of the vacuum check and the vacuum diaphragm which are not a part of the design of Fig. 170. The vacuum-check disc at the vent port prevents the ready return of air into the valve. A slight vacuum produced in the air valve causes the external air pressure, exerted through the atmospheric port upon the vacuum diaphragm, to force the float to be lifted and to make its needle to positively close the seat opening. Valves similar to that shown in Fig. 171 prevent the return of air to a radiator and allow a vacuum to form in the heating system as the result of steam condensation so that vapor may be supplied to the radiators at a temperature less than



(*Hoffman Specialty Co.*)

FIG. 171. Vacuum radiator air valve.



(*Hoffman Specialty Co.*)

FIG. 172. Main-vent automatic air valves.

212 F when the fire in the boiler furnace is not active. Under such conditions heat will be maintained in the radiators for a longer period of time than when a vacuum is not created in the system.

Air valves embodying similar features and suitable for the ends of one-pipe and two-pipe steam mains are shown by Fig. 172. In the illustration of the main vent (vacuum) the valve is shown to function when the barometric pressure exerted through the atmospheric tube within the bellows holds the float in a position so that its pin closes the vent port. The main vent (open) is primarily a valve for air elimination from the piping of steam unit heaters, but it may be satisfactorily used at the ends of steam mains.

223. Flow of Steam in Pipes. Frictional resistance is offered to the flow of fluids by the interior surfaces of pipes. The resistances are overcome by the expenditure of a part of the pressure which the fluid initially exerts upon the pipe walls. In a pipe line L feet in length, when the initial fluid pressure is p_1 at one end and the final fluid pressure on the pipe walls is p_2 after the fluid has traveled L feet, the fluid pressure loss is $p = p_1 - p_2$, all pressures being in pounds per square inch.

The frictional resistance to flow is independent of the absolute pressure exerted by the fluid on the pipe walls; it varies almost as the square of the velocity of flow and directly as the density of the fluid, the pipe length, and the interior perimeter of the pipe.

Various equations have been developed for use in the calculation of the pressure losses of fluids as they flow in ducts and pipes. Chief among these are those (1) which employ a dimensionless ratio known as Reynold's number, and (2) those which express pressure losses in feet head of fluid flowing as a function of $v^2/2g$, where v is the velocity of flow, fps; and g is 32.17 ft per sec^2. For the flow of steam in pipes Babcock's equation, based on the theory underlying the second designated method, is satisfactory for most purposes.

The loss of head, in feet of fluid flowing, is $h = 144p/d$, where p is the pressure loss, psig in a given length of pipe; and the fluid density is d lb per cu ft. A commonly used equation for the losses of head in pipes and ducts is

$$h = \frac{fLRv^2}{A2g} \tag{101}$$

here h = loss of head in feet of fluid flowing.
f = a dimensionless coefficient of friction.
L = duct or pipe length, ft.
R = duct or pipe circumference, ft.

v = velocity of flow, fps.
A = duct or pipe internal cross-sectional area, sq ft.
g = acceleration due to gravity, 32.17 ft per sec^2.

The values of R and A, in equation 101, can be expressed in terms of the pipe diameter D ft as $R = \pi D$ and $A = \pi D^2/4$ to give

$$h = \frac{f 2 L v^2}{D g} \tag{102}$$

Unwin expressed the coefficient f as $K(1 + 3/10D)$ which becomes $K(1 + 3.6/D_1)$ where D_1 is the internal diameter of the pipe in inches. G. H. Babcock experimentally established $K = 0.0027$ for the flow of steam in pipes. When the pipe diameter is known the weight of steam flowing per minute with a velocity of v fps can be calculated. The average density d of the steam is the reciprocal of its average specific volume v_g or $1/v_g$. Also the velocity of flow in feet per second may be computed when the weight of steam flowing per minute is known. Derived equations for w and v are

$$w = 60v\frac{\pi}{4}\left(\frac{D_1}{12}\right)^2\frac{1}{v_g} \quad \text{and} \quad v = \frac{9.6wv_g}{\pi D_1{}^2}$$

When use is made of the preceding expressions and relations Babcock's equation may be developed from equation 102 as

$$p = 0.0001306\left(1 + \frac{3.6}{D_1}\right)\frac{w^2 L v_g}{D_1{}^5} \tag{103}$$

The portion of equation 103, represented by $0.0001306\ (1 + 3.6/D_1) \div D_1{}^5$ always remains a constant for a given size of pipe and can be represented by a symbol C in the following equation

$$p = Cw^2 L v_g \tag{104}$$

Values of C for commercial steel and wrought-iron pipe are given in Table 74.

224. Allowances for Pipe Fittings and Entrance Losses. The pressure loss of steam flowing in a pipe as given by equation 104 is based on a straight length of pipe equal to L feet. Additional frictional resistance is also offered as steam must enter the pipe through a square-edged opening such as the outlet connection from the steam space of a boiler. Pipe fittings like tees, elbows, and valves also offer additional resistances to flow. A common practice is to make allowances for additional resistances offered by valves and fittings. This is done by estimating the length of straight pipe which has the frictional

TABLE 74

Constant *C* for Babcock's Equation

Nominal Iron Pipe Size, In.	Actual Inside Diameter of Pipe, In.	Constant *C*	Nominal Iron Pipe Size, In.	Actual Inside Diameter of Pipe, In.	Constant *C*
1	1.049	0.000,456	4	4.026	0.000,000,234
1¼	1.380	0.000,094,1	5	5.047	0.000,000,068,3
1½	1.610	0.000,039,1	6	6.065	0.000,000,024,1
2	2.067	0.000,009,48	8	7.981	0.000,000,005,85
2½	2.469	0.000,003,50	10	10 020	0.000,000,001,76
3	3.068	0.000,001,01	12	12.000	0.000,000,000,682
3½	3.548	0.000,000,468			

equivalent of each pipe entrance and fitting. The extra allowance for each additional resistance is added to the length of straight pipe in the line, and the value of L in equation 104 should be the equivalent length of the line. Table 75 gives equivalents which may be used in computing the value of L.

TABLE 75

Length in Feet of Pipe to be Added to Actual Length of Run—Owing to Fittings—to Obtain Equivalent Length*

Size of Pipe, In.	90-Deg Elbow	Side Outlet Tee	Gate Valve†	Globe Valve†	Angle Valve†
	Length in Feet to Be Added to Run				
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	155

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* From *ASHVE Guide*, 1948.

† Valve fully open.

In order to estimate heat losses from pipe fittings their external areas must be known. These depend upon the design of the fittings, but the areas of Table 76 may be taken as representative.

TABLE 76*

AREAS OF FLANGED FITTINGS, SQUARE FEET†

Nominal Pipe Size, In.	Flanged Coupling		90-Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

* *ASHVE Guide*, 1948.

† Including areas of accompanying flanges bolted to the fitting.

225. Example of the Calculation of the Pressure Loss in a Steam Line. The following will serve as an example of the calculation of the pressure losses in a low-pressure steam line.

Example. A 6-in. steam line receives steam initially dry at a pressure of 20 psia and must deliver the equivalent of 5000 lb of dry steam per hour at the outlet end. The line has 400 ft of straight pipe, one gate valve, 25 pairs of flanges, and six 90-deg elbows. The pipe is insulated with covering which is 80 per cent efficient and is installed in a conduit where the average air temperature is 95 F. The loss of heat per square foot of bare pipe surface may be taken as 2.2 Btu per sq ft per hr per deg F. Find the pressure losses in pounds per square inch.

Solution. The external areas of the bare pipe and fittings are from Tables 68 and 76.

Straight pipe, 400 × 1/0.576	=	694.4 sq ft
Gate valve, same as a tee, 1 × 7.8	=	7.8
Flanges, 25 × 1.82	=	45.5
Elbows, 6 × 5.13	=	30.8
Total		778.5 sq ft

The weight of dry steam condensed in the line at 20 psia is

$$778.5 \times \frac{2.2(228 - 95)(1 - 0.8)}{960.1} = 47.4 \text{ lb per hr}$$

When the average steam-line temperature is taken as 228 F the enthalpy of evaporation is 960.1 Btu per lb. These values are taken with small error for a pressure of 20 psia as the average steam pressure in the line is not known at this stage of the calculations.

The total dry steam which must be fed into the line is 5000 + 47.4 = 5047.4 lb per hr or 5047.4 ÷ 60 = 84.1 lb per min. At 20 psia pressure, $v_g = 20.089$ cu ft per lb. The equivalent length L of the line, Table 75 is 400 + (6 × 13) + 2.8 = 480.8 ft. Then $p = 0.000{,}000{,}0241 \times 84.1^2 \times 480.8 \times 20.089 = 1.65$ psig. This value is sufficient for most purposes. If greater accuracy is desired, the approximate pressure loss, as just computed, being known, the calculation can be remade using 20 − 0.83 = 19.17 psia as the average pressure in the steam line. The pressure loss as computed is (100 × 1.65) ÷ 480.8 = 0.343 psig or 5.5 oz per 100 equivalent feet of length.

A pressure loss of one ounce or 0.0625 psig. per 100 equivalent ft is considered satisfactory in designing the steam-distributing piping of many steam-heating systems.

226. Steam-Flow Pressure-Loss Charts. The data relative to the flow of steam in commercial pipes, as obtained by the use of Babcock's equation, are often presented graphically as in Fig. 173. The data are affected by both the average pressure and the condition of the steam. The data of Fig. 173 are true for dry saturated steam at 16 psia.

227. The Piping Design of Steam-Heating Systems. The pipes of steam-heating systems are obliged at times to carry three media: steam, air, and water. The condition often arises where water must flow in a pipe in a direction counter to the steam flow. All pipes must have capacity to function under heavy load conditions, and provisions must be made for expansion, contraction, drainage of water, and the removal of air and other non-condensable gases.

The sizes of pipe based on the allowable pressure losses may be determined by the method of Art. 223 when the flow of steam and water is parallel as in mains and down-feed risers. Because of the possibility of the critical velocity of steam flow being reached or exceeded in pipes where counterflow of steam and water occur, this method should not be used unless the pipe size is based on an allowable steam velocity which is below the critical velocity of flow. The critical velocity is the one at which steam will pick up and carry with it water flowing counter to the steam flow. Places where critical velocity of flow should be considered are in the branches from mains to undripped up-feed risers and the runouts from risers to one-pipe steam radiators. A partially closed valve of a one-pipe steam radiator will also cause the critical velocity of flow to be exceeded at that point. In two-pipe up-feed supply risers the counter-flowing condensation is generally not sufficient to cause trouble if the risers are insulated as they should be. Critical velocities of steam flow are variable depending upon the size of the pipe, its pitch, and the amount of water handled. They range from 11 to 70 fps. Generally, critical velocities are lower in small pipes than in larger ones.

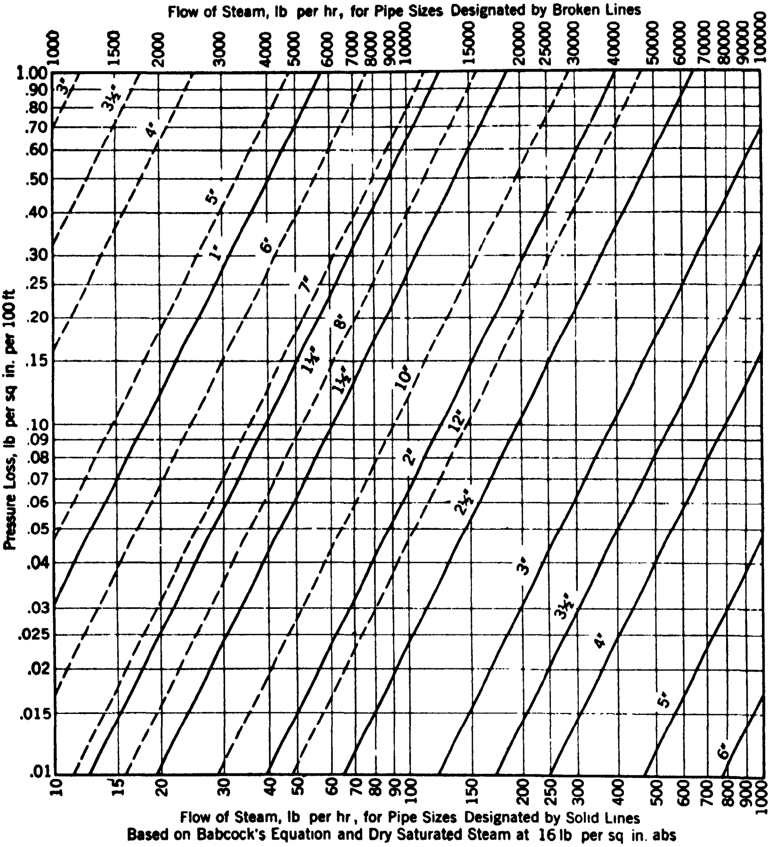

FIG. 173. Pressure losses and flow of steam in commercial pipes.

Whenever possible, pipe sizes may be determined by means of either Babcock's equation or a chart similar to that of Fig. 173. The total equivalent length of pipe cannot be determined for Babcock's equation

TABLE 77*

STEAM-PIPE CAPACITIES FOR LOW-PRESSURE SYSTEMS†·‡

Reference to this table will be by column letter *A* through *L*

Pipe Size, In.	Capacities of Steam Mains and Risers								Special Capacities for One-Pipe Systems Only		
	Direction of Condensation Flow in Pipe Line										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam, Two-Pipe Only		Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
	$\frac{1}{32}$ Psi or $\frac{1}{2}$-Oz Drop	$\frac{1}{24}$ Psi or $\frac{2}{3}$-Oz Drop	$\frac{1}{16}$ Psi or 1-Oz Drop	$\frac{1}{8}$ Psi or 2-Oz Drop	$\frac{1}{4}$ Psi or 4-Oz Drop	$\frac{1}{2}$ Psi or 8-Oz Drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*§	*I*‖	*J*¶	*K*	*L*‖
	Capacity Expressed, Sq Ft Edr										
¾	...	...	30	...	...	...	30	...	25	...	...
1	39	46	56	79	111	157	56	34	45	28	28
1¼	87	100	122	173	245	346	122	75	98	62	62
1½	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2½	449	518	635	898	1,270	1,800	635	395	464	...	260
3	822	948	1,160	1,650	2,330	3,290	1130	700	800	...	475
3½	1,230	1,420	1,740	2,460	3,470	4,910	1550	1150	1140	...	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2040	1700	1520	...	1110
5	3,210	3,710	4,550	6,430	9,090	12,900	...	3150	...	...	2180
6	5,280	6,100	7,460	10,550	14,900	21,100	...	...	...	...	...
8	11,000	12,700	15,500	21,970	31,070	43,900	...	...	...	...	...
10	20,000	23,100	28,300	40,100	56,700	80,200	...	...	...	...	...
12	32,000	37,100	45,500	64,300	91,000	129,000	...	...	...	...	...
16	61,000	69,700	84,800	121,000	170,000	242,000	...	...	...	...	...
	Capacity Expressed, Lb per Hr										
¾	...	...	8	...	...	...	8	...	6	...	7
1	10	12	14	20	28	40	14	9	11	7	7
1¼	22	25	31	43	61	87	31	19	20	16	16
1½	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2½	112	130	159	225	318	449	159	99	116	...	65
3	206	237	291	411	581	822	282	175	200	...	119
3½	307	355	434	614	869	1,230	387	288	286	...	186
4	435	503	614	869	1,230	1,740	511	425	380	...	278
5	806	928	1,140	1,610	2,270	3,210	...	788	...	...	545
6	1,320	1,520	1,870	2,640	3,730	5,280	...	...	...	...	...
8	2,750	3,170	3,880	5,490	7,770	11,000	...	...	...	...	...
10	5,010	5,790	7,090	10,000	14,200	20,000	...	...	...	...	...
12	8,040	9,290	11,400	16,100	22,700	32,200	...	...	...	...	...
16	15,100	17,400	21,200	30,300	42,400	60,500	...	...	...	...	...
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

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* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

† This table is based on pipe-size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

‡ All drops shown are in pounds per square inch per 100 ft of equivalent run—based on pipe properly reamed. Steam at an average pressure of 1 psi is used as a basis for calculating capacities.

§ Do not use column *H* for drops of $\frac{1}{24}$ or $\frac{1}{32}$ psi; substitute column *B* or column *C* as required.

‖ On radiator runouts over 8 ft long, increase one pipe size over that shown.

¶ Do not use column *J* for drop of $\frac{1}{32}$ psi except on sizes 3 in. and over; below 3 in. substitute column *B*.

TABLE 78

RETURN-PIPE CAPACITIES*

Reference to this table will be by column letter *M* through *EE*

Pipe Size, In.	Capacity of Return Mains and Risers†																
	Mains																
	$\frac{1}{32}$-Lb or $\frac{1}{2}$-Oz Drop per 100 Ft			$\frac{1}{24}$-Lb or $\frac{2}{3}$-Oz Drop per 100 Ft			$\frac{1}{16}$-Lb or 1-Oz Drop per 100 Ft			$\frac{1}{8}$-Lb or 2-Oz Drop per 100 Ft			$\frac{1}{4}$-Lb or 4-Oz Drop per 100 Ft			$\frac{1}{2}$-Lb or 8-Oz Drop per 100 Ft		
	Wet	Dry	Vacuum	Wet	Dry	Vacuum	Wet	Dry	Vacuum	Wet	Dry	Vacuum	Wet	Dry	Vacuum	Wet	Dry	Vacuum
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
¾	...	...	...	...	...	326	...	...	400	...	...	568	...	...	800	...	...	1,130
1	500	248	...	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	...	...	1,980
1¼	850	520	...	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	...	...	3,390
1½	1,350	822	...	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	...	...	5,370
2	2,800	1,880	...	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	...	...	11,300
2½	4,700	3,040	...	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	...	...	18,900
3	7,500	5,840	...	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	...	...	30,200
3½	11,000	7,880	...	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	...	...	45,200
4	15,500	11,700	...	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	...	...	62,190
5	...	...	...	...	...	31,500	...	...	38,700	...	...	54,900	...	...	77,400	...	...	109,000
6	...	...	...	...	...	50,450	...	...	62,000	...	...	88,000	...	...	124,000	...	...	175,000
	Risers																	
¾	...	190	...	...	190	570	...	190	700	...	190	994	...	190	1,400	...	...	1,980
1	...	450	...	...	450	976	...	450	1,200	...	450	1,700	...	450	2,400	...	...	3,390
1¼	...	990	...	...	990	1,550	...	990	1,900	...	990	2,700	...	990	3,800	...	...	5,370
1½	...	1500	...	...	1500	3,260	...	1500	4,000	...	1500	5,680	...	1500	8,000	...	...	11,300
2	...	3000	...	...	3000	5,450	...	3000	6,700	...	3000	9,510	...	3000	13,400	...	...	18,900
2½	...	...	...	...	...	8,710	...	...	10,700	...	...	15,200	...	...	21,400	...	...	30,200
3	...	...	...	...	...	13,000	...	...	16,000	...	...	22,700	...	...	32,000	...	...	45,200
3½	...	...	...	...	...	17,900	...	...	22,000	...	...	31,200	...	...	44,000	...	...	62,200
4	...	...	...	...	...	31,500	...	...	38,700	...	...	54,900	...	...	77,400	...	...	109,000
5	...	...	...	...	...	50,500	...	...	62,000	...	...	88,000	...	...	124,000	...	...	175,000

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[*] This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers. Used by permission.

[†] Capacity expressed in square feet of equivalent direct radiation.

until the pipe size is established by a trial calculation using only the total length of straight pipe involved. After the trial size is determined, the calculation for actual pressure losses is made using an equivalent length of pipe, which includes the allowances for fittings, and if the computed pressure losses per 100 ft do not exceed those specified or desirable the sizing is completed. If the pressure losses are excessive the next larger size of pipe must be taken and the computations made again.

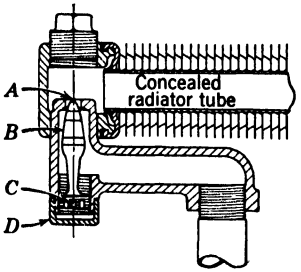
The pressure losses allowable in heating systems vary somewhat with the system. In general, they should not exceed 1 oz per 100 ft, and it may be desirable to have them less.

228. Tables of Steam-Heating Pipe Sizes. As a result of extensive research, conducted through the agency of the American Society of Heating and Ventilating Engineers, pipe sizes to serve various amounts of standard direct radiation have been ascertained for the various types of steam heating systems. These data are presented in Tables 77 and 78, which are reproduced by special permission.

229. Orifices for Steam Radiators and Convectors. The distribution of steam to the various radiators and convectors of a heating system is of prime importance. Each heat disseminator should receive simultaneously with the others of the system an adequate amount of steam in accordance with the supply pressure and the heat requirements of the space at its locations in order to give uniformity of heating and satisfactory air temperatures. Commercial sizes of pipes and valves, based on the data of Table 77, operate with variations of steam pressures and volumes for reasons mentioned in Art. 223. Some form of a radiator orifice may be used to balance a two-pipe system; its use causes operating difficulties in one-pipe systems because of interference with water drainage from the heat disseminators installed.

Orifices available are those having adjustable openings and those with fixed ports. An adjustable orifice may be part of the radiator control valve or it may be a separate fitting. Adjustable orifices have the advantage over fixed-opening units in that they do not have the limited range of capacity that a particular size of fixed-opening unit has. Although orifices can be placed at various locations in the steam piping of a heating system to restrict flow the most convenient and satisfactory placement is at the steam inlet of each heat disseminator.

The offset-type adjustable orifice shown by Fig. 174 is applicable to installation with certain forms of convectors and radiators. The port *A*, through which steam enters the concealed radiator, is higher than the lower part of the radiator tube so that water cannot accumulate above it and cause noise. The conical-shaped restrictor *B* is adjustable for a radiator of any size and stack height by removing water- and steam-tight cap *D* and rotating screw *C* by means of a screwdriver. Figure 175*a* shows an angle-type radiator valve with a union and a tail piece or nipple which is screwed into the radiator tapping bushing. A fixed-orifice plate may be placed within the nipple as shown by *b* or it may be located in the union at *c* in Fig. 175. As a result of investigations by Sandford and Sprenger[1] the sizes of orifices as given by Table 79 are recommended for different amounts



(*C. A. Dunham Co.*)

FIG. 174. Section through adjustable regulating fitting for concealed radiation.

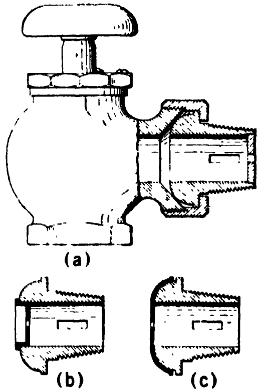


FIG. 175. Metering orifice.

TABLE 79

RADIATOR ORIFICE SIZES

Based on orifice plates $\frac{3}{32}$ in. thick with a maximum steam pressure differential of 6 in. of mercury

Radiator Rating, Sq Ft	Orifice Diameter, In.	Radiator Rating, Sq Ft	Orifice Diameter, In.	Radiator Rating, Sq Ft	Orifice Diameter, In.
18	$\frac{3}{32}$	44	$\frac{5}{32}$	80	$\frac{7}{32}$
24	$\frac{7}{64}$	52	$\frac{11}{64}$	91	$\frac{15}{64}$
30	$\frac{1}{8}$	61	$\frac{3}{16}$	102	$1\frac{1}{4}$
37	$\frac{9}{64}$	70	$\frac{13}{64}$		

---

[1] "Flow of Steam through Orifices into Radiators," by S. S. Sandford and C. B. Sprenger, *ASHVE Trans.*, Vol. 37, 1931.

of radiation. These data are based on a maximum drop in steam pressure of 6 in. of mercury through an orifice plate $\frac{3}{32}$ in. thick.

The orifice system is inexpensive to install, but it requires considerable attention in the matter of steam-pressure adjustment if the maximum savings in the use of steam are to be made. Table 80 gives the differential pressures across the orifice plates recommended by Sanford and Sprenger when the outdoor air temperatures vary.

TABLE 80

NECESSARY ORIFICE STEAM-PRESSURE DIFFERENTIAL FOR SEVERAL OUTSIDE-AIR TEMPERATURES

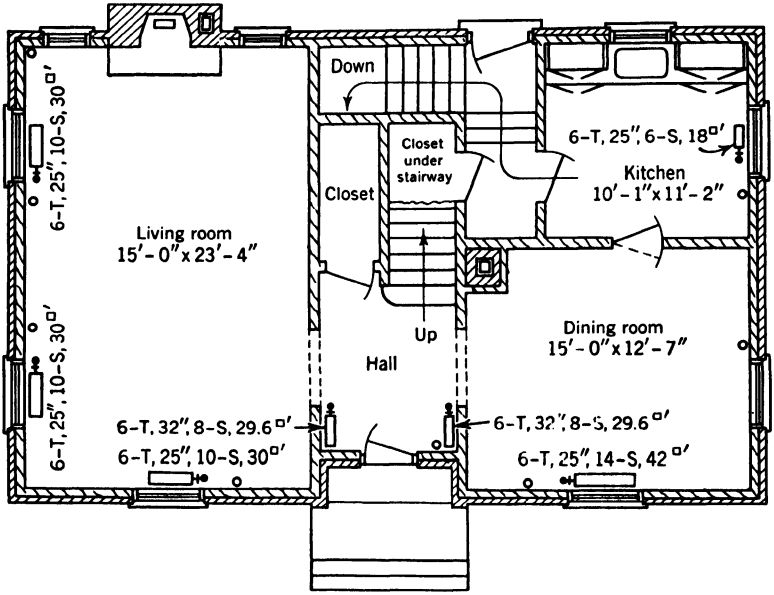
Room-air temperature 70 F; steam pressure in radiator 0 psig

Outside-Air Temperature, Deg F	Pressure Differential, In. of Mercury	Outside Air Temperature, Deg F	Pressure Differential, In. of Mercury
0	6	40	1.2
10	4.5	50	0.5
20	3.2	60	0.1
30	2.1		

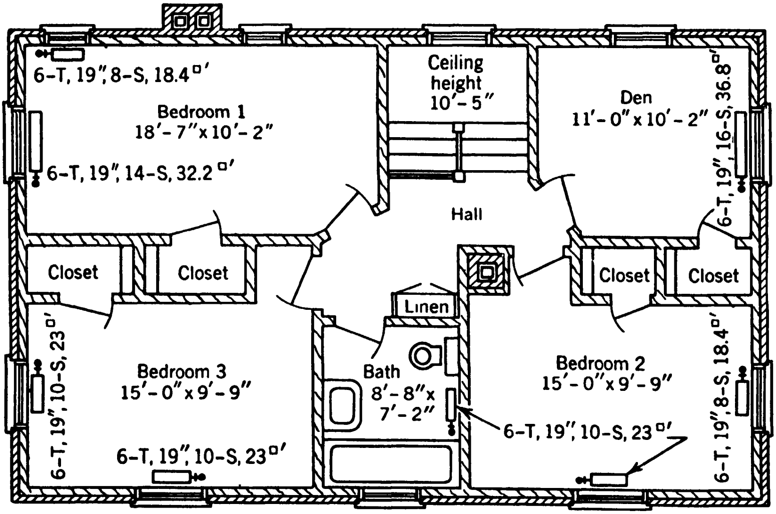
230. Heating Plans. Working drawings of proposed heating jobs are essential and should give with clarity the necessary information. Symbols for pipes, fittings, and other pieces of equipment are given in Table 134 of the Appendix. Use of these symbols saves time and prevents the drawing from becoming difficult to read because of a multiplicity of detail lines.

Figures 176 and 177 illustrate the use of symbols in the layout of heating plans. The scheme of placing the risers on the drawing as shown by Fig. 177 is satisfactory for small layouts. When a large number of radiators are to be served by a riser the scheme of Fig. 178 is better. Here the various risers are numbered on the floor plans, and separate details, such as Fig. 178, are placed at convenient places on the plan sheets to show the risers, their sizes at various points, and the amounts of radiation to be served.

231. Piping Details. The statements of Art. 207, relative to piping connections at boiler outlets are true for both steam and hot-water boilers. Figure 177 indicates how the plan of the piping around a steam boiler may appear. In addition many other details are necessary for the success of various steam heating systems. Figure 179 shows some of the more common arrangements for both one-pipe and vapor and vacuum steam systems. These details are worked out with the idea of providing for expansion, the avoidance of water pockets which cannot be drained, the drainage of the system, and the removal of air from the supply mains and radiators.



(a) First Floor



(b) Second Floor

Fig. 176. Radiator sizes and locations. One-pipe steam heating system for a residence.

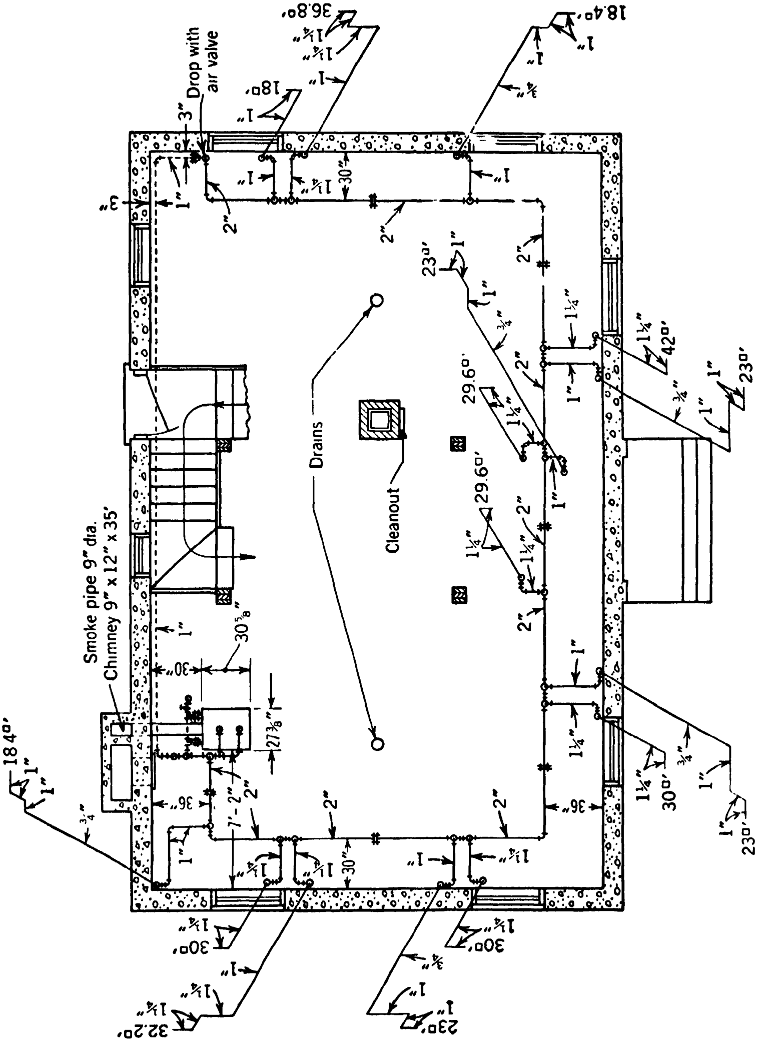


FIG. 177. Basement plan of a one-pipe steam heating system for a residence.

232. District Heating with Steam. Frequently in cities and about institutions a single boiler plant to furnish heat to buildings is more feasible and economical than having boiler equipment located in each building. The vehicle used to carry the heat to the buildings may be either steam or water. Water, although used to some extent, is not as easily handled as steam as it must be pumped. The problems of metering hot water and the pressures required in tall buildings also present some difficulties.

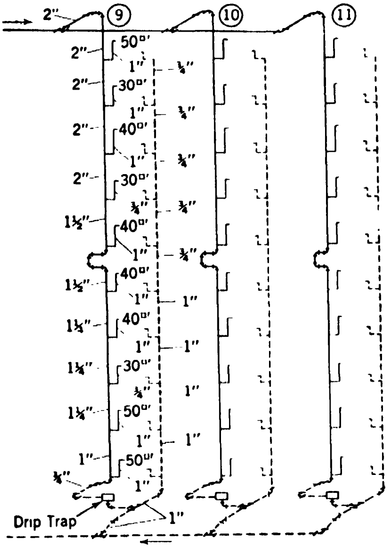


FIG. 178. Riser diagram for a two-pipe steam heating system.

Steam may be generated in large modern boiler plants with an efficiency of fuel utilization ranging from 70 to 88 per cent. Although losses occur in the distributing mains the final overall efficiency of fuel-heat utilization, taking into account all losses from fuel supply to heated spaces, ranges from 60 to 65 per cent in district heating systems. With small individual boiler plants the yearly average overall plant efficiency may average from 50 to 60 per cent.

The steam delivered by the boiler plant may arrive at the place of consumption at either high or low pressure. Often the steam pressure has to be reduced at the building where it is used. This can be done by pressure-reducing valves, or, where it is desired and the steam pressure is great enough, the pressure reduction can be effected by means of steam engines and turbines. These units serve as rotary pressure-reducing valves and produce power which has a commercial value along with the exhaust steam available for heating. The heating system used within the building may be of the types previously described.

233. Steam-Distributing Lines in District Heating. Generally where large areas are served the mains are underground. These mains carry steam, at pressures ranging from 5 psig and up, and supply service through a system of smaller pipes to all points where steam is required. The service connections to individual buildings are taken from the underground distributing system.

Steam is transported from the boilers by means of either trunk or

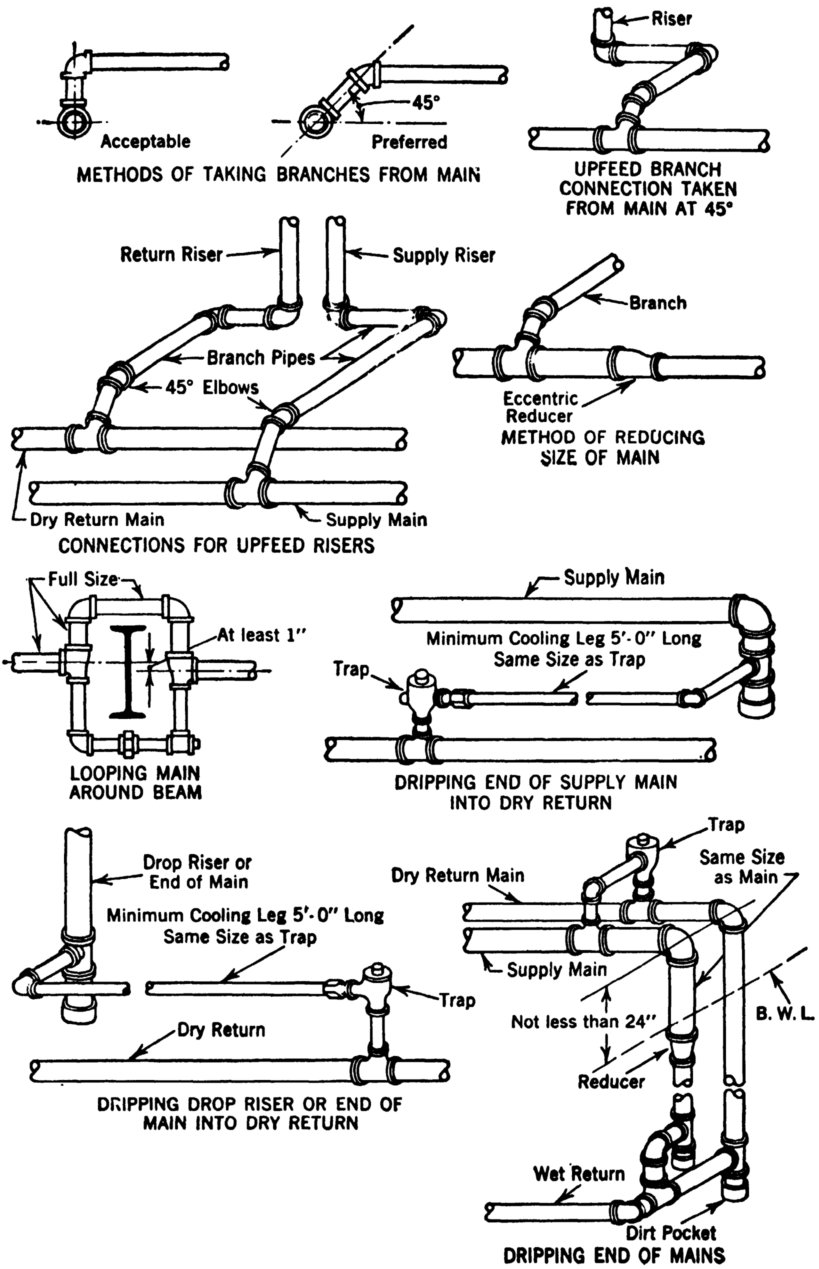

Acceptable
45°
Preferred
METHODS OF TAKING BRANCHES FROM MAIN
Riser
UPFEED BRANCH CONNECTION TAKEN FROM MAIN AT 45°
Return Riser
Supply Riser
Branch Pipes
45° Elbows
Dry Return Main
Supply Main
CONNECTIONS FOR UPFEED RISERS
Branch
Eccentric Reducer
METHOD OF REDUCING SIZE OF MAIN
Full Size
At least 1"
LOOPING MAIN AROUND BEAM
Supply Main
Minimum Cooling Leg 5'-0" Long Same Size as Trap
Trap
DRIPPING END OF SUPPLY MAIN INTO DRY RETURN
Drop Riser or End of Main
Minimum Cooling Leg 5'-0" Long Same Size as Trap
Trap
Dry Return
DRIPPING DROP RISER OR END OF MAIN INTO DRY RETURN
Trap
Dry Return Main
Same Size as Main
Supply Main
Not less than 24"
B. W. L.
Reducer
Wet Return
Dirt Pocket
DRIPPING END OF MAINS

FIG. 179. Steam heating piping details.

feeder mains. Trunk mains leave the boiler plant with their maximum size and decrease in size as lateral distributing pipes are taken from them. Feeder mains of uniform size carry steam at high velocities and often at high pressures to important locations where it is delivered to the distributing system at the proper pressure. The feeder system has the advantage that smaller lines leaving the plant may be used than are possible with the more prevalent trunk system. District-heating pipe sizes may be computed from Babcock's formula, equation 104.

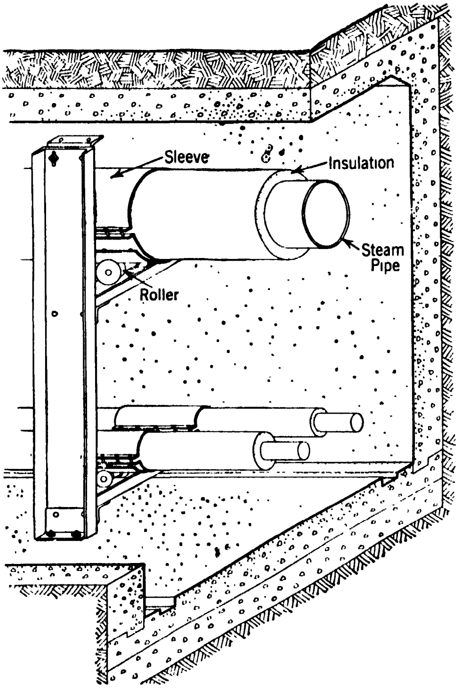


FIG. 180. Tunnel for underground steam mains at the University of Illinois.

234. Underground Tunnels and Conduits. Steam pipes placed underground must be protected from mechanical damage, corrosion, moisture, and the loss of heat from them; also pipe drainage and provision for expansion must be given attention.

Underground steam pipes are placed in tunnels or pipe conduits; the conduits have many forms. Wherever possible, tunnels, Fig. 180, are very desirable as the pipes are more accessible and the pipe insulation can be kept dry. The tunnel may also serve to carry telephone and power cables as well as the lines for returning the condensate to the boiler house. Figure 181 shows the details of the methods of anchoring both the steam pipe and its expansion joints in the tunnel of Fig. 180.

The pipes are supported upon rollers with pipe insulation between the rollers and the mains. The insulation between the pipe and its roller support carries the weight of the pipe. The outer surface of the "85 per cent magnesia" covering of the pipes is protected from abrasive action by the pipe roller by sheet-metal sleeves placed about the covering at the roller locations.

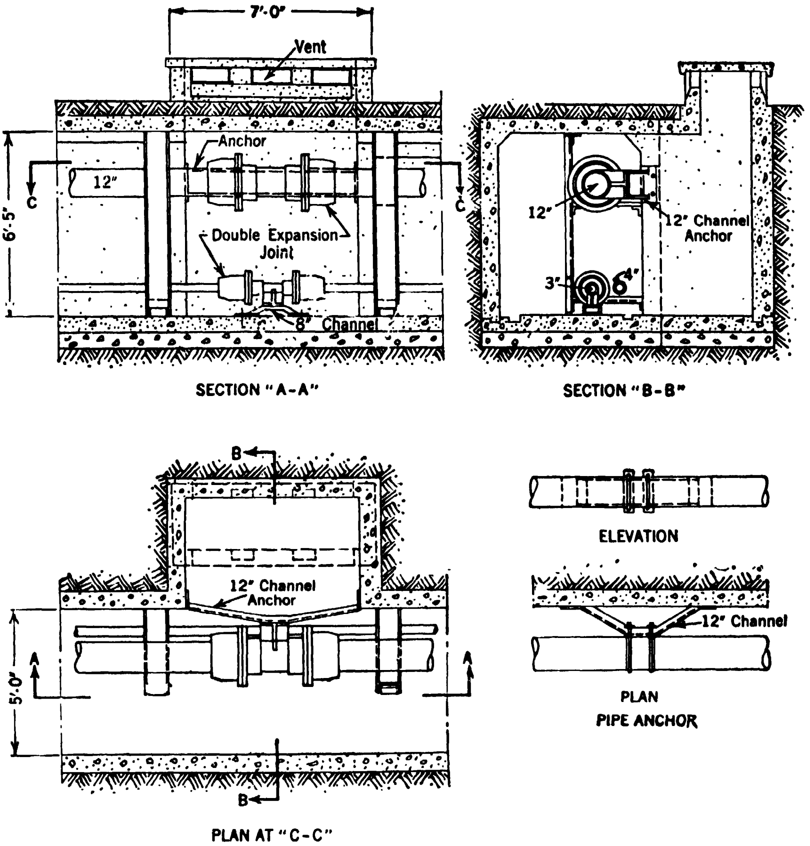


Fig. 181. Expansion-joint and pipe-anchor details, University of Illinois.

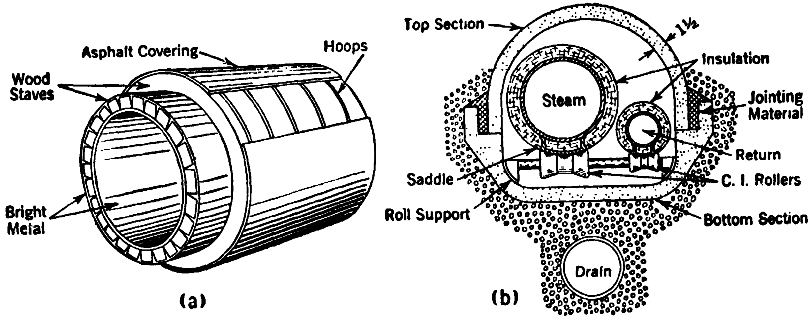
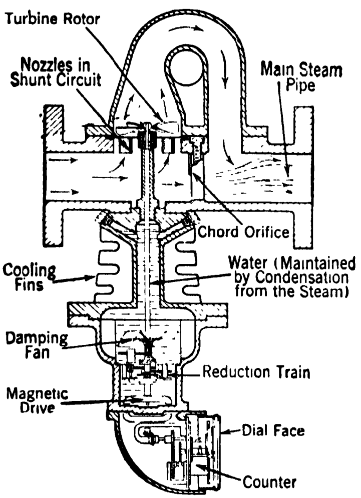


Fig. 182. (a) Wood-stave pipe conduit. (b) Precast concrete pipe conduit.

Two forms of pipe conduit, which are buried in the ground, are illustrated by Fig. 182. Conduit *a*, made in sections with tongue-and-groove joints at the ends, has wood staves bound together with wire hoops and is coated on the outside with an asphalt material as a protection against moisture from the earth. Inside the conduit is a tin-plated sheet-metal lining separated from the wooden walls by an air space. The pipe is placed within the metal lining without insulation other than that provided by the wood, the metal lining, and the air space. Conduit *b* is comprised of split sections of tile cemented together at tongue-and-groove joints. The pipes are supported on rollers properly located, and loose insulation may be packed within the conduit around the pipes.

Manholes must be constructed for conduits at intervals along the line to house stop valves, drainage traps, and expansion joints.

235. Other Details of District Heating. The steam pipes in tunnels and conduits must be properly supported and also anchored at various points. Expansion may be provided for by packed and packless expansion joints, pipe bends, and loops. The flow of steam and condensation within the pipes should be in the same direction wherever possible, and drip traps are necessary to remove condensation at low places in the lines. Good insulation, which can be kept dry, should be used to cover underground steam pipes and condensation returns. Tunnels and conduits should be buried deep enough to secure the maximum insulating effect of the surrounding earth. Whenever possible the condensation from the distributing lines and that from the buildings served should be returned to the boilers to effect a saving of both heat and water. These return lines may be carried in the tunnels and conduits. Stop valves are required at strategic locations so that the steam may be shut off when repairs to the distributing lines are necessary.

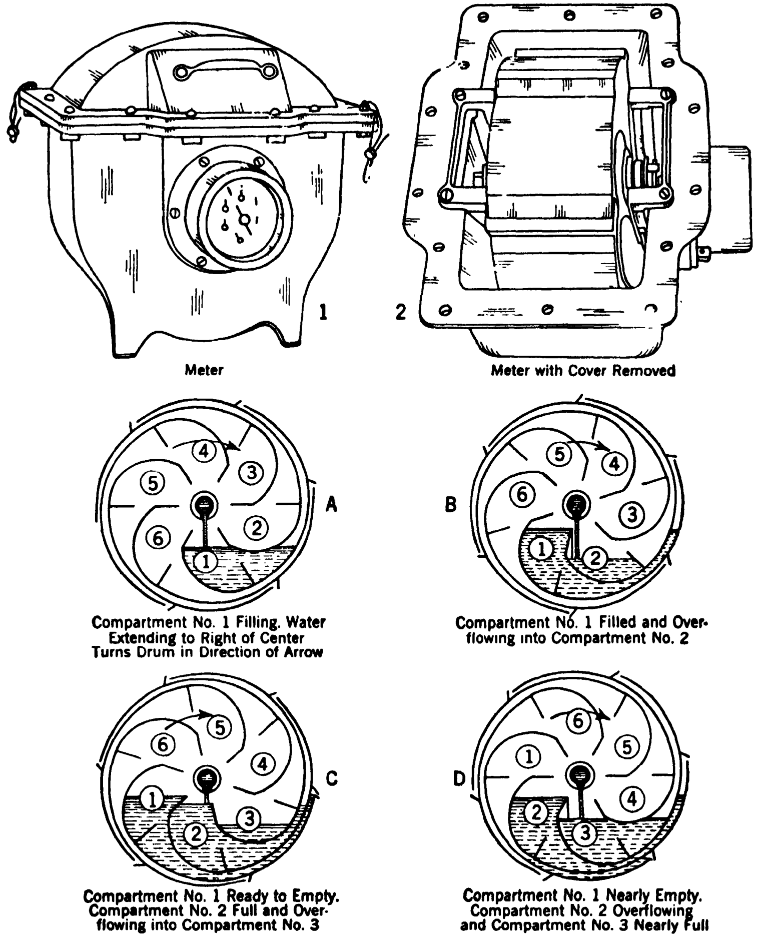


(*Builders Iron Foundry*)

FIG. 183. Sectional view of a shunt steam meter.

The heating systems served should be susceptible to and fitted with the proper temperature control of the air of the building; otherwise they may use an excessive amount of steam.

236. Measurement of Steam Used. Steam sold to consumers by district heating companies may be retailed on either a flat rate basis or on a rate per 1000 lb of steam used. In any event, it is desirable to know the amount of steam used in a building.



Meter

Meter with Cover Removed

Compartment No. 1 Filling. Water Extending to Right of Center Turns Drum in Direction of Arrow

Compartment No. 1 Filled and Overflowing into Compartment No. 2

Compartment No. 1 Ready to Empty. Compartment No. 2 Full and Overflowing into Compartment No. 3

Compartment No. 1 Nearly Empty. Compartment No. 2 Overflowing and Compartment No. 3 Nearly Full

(*Central Station Steam Co.*)

FIG. 184. Condensation meter.

The steam can be measured, as it enters the building, with a meter. Several types of meter are available; Fig. 183 illustrates one form. This meter operates on the basis that the steam flow is proportional to the pressure differential produced by an orifice placed in the meter. The pressure difference produced causes a flow of steam through the nozzles to revolve the turbine blades in the shunt circuit. The turbine rotor drives the steam-flow recording mechanism.

Steam consumption is also measured by condensation meters; Fig. 184 illustrates the appearance and action of one type. It has a cylindrical copper drum which revolves on roller bearings within the cast-iron case. The condensate is introduced into the drum at its center by a slotted spout which does not touch the drum. The drum has scroll-shaped compartments, and as each compartment fills with water the center of gravity of the water shifts, causing the drum to rotate. Each compartment discharges a definite quantity of water each revolution of the drum. The motion of the drum is transmitted to a counter attached to the meter case. Although the meter compartments measure the water volumetrically the record of the counter is in pounds of condensate passed through the meter.

PROBLEMS

1. A 4-in. steam pipe is to carry steam with a pressure loss not to exceed 1.5 oz per 100 ft of equivalent length. The equivalent length of the line is 300 ft, and the initial pressure of the steam is 5.5 psig when the barometric pressure is 14.5 psi. If the heat losses from the line are neglected, find the weight of steam that it will handle in pounds per hour.

2. A pipe line is to have 500 ft of straight length, 2 gate valves, 28 pairs of flanges, and five 90-deg elbows. The line is to be designed to deliver 9000 lb of steam per hr. The initial pressure of the steam is to be 25 psia, and the total pressure loss is not to exceed 2 psi when the pipe is covered with 1-in. air-cell covering and the temperature of the air surrounding the installation is 90 F. Find the required size of pipe.

CHAPTER 11

HEATING WITH HOT WATER

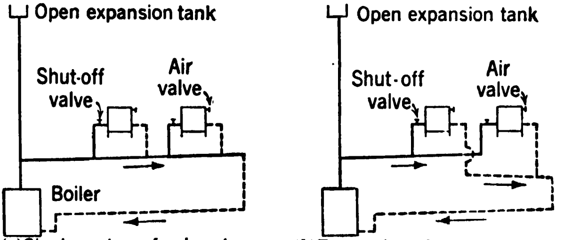
237. Applications and Classifications of Systems. The use of water as a heat-carrying medium finds applications in various forms of systems which serve structures such as small residences and buildings covering considerable ground area and, in some cases, of considerable height. Water has the advantage over steam that its temperature may be easily regulated according to weather conditions so that air-temperature control within rooms is thereby facilitated.

Flow of water in a system may be produced as the result of either density differences (see Art. 248) or by the action of a pump. On these bases, system classifications are gravity flow and forced circulation. Open systems have atmospheric pressure exerted upon the water of an expansion tank. Pressure systems may or may not have an expansion tank and function with much higher water pressures and temperatures than are possible with open systems.

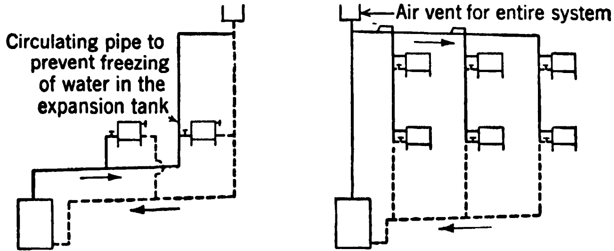
Open and pressure systems may have either up-feed or down-feed arrangements of the supply risers. Up-feed systems are further separated into (1) one-main installations which have a single pipe to serve as both a supply and return main, and (2) two-main plants having separate supply and return mains. Overhead down-feed systems have supply mains above the level of the highest radiators served and return mains below the lowest radiators. All overhead systems may have either single or double down-feed risers. The overhead arrangement is prone to give more positive gravity circulation of water than is possible with up-feed risers. Examples of the foregoing forms of hot-water systems are illustrated by Fig. 185.

238. Single-Main Gravity Up-Feed System. The arrangement of the mains and risers in this system, illustrated in Fig. 185*a*, is the same as in the one-pipe forced-circulation system described later in Art. 257, except that the main size is not reduced between the two tees which connect each radiator to it. Because of the necessity of reducing friction loss to a minimum all pipe sizes are proportionately larger in the gravity system. The perfection of the one-pipe forced-circulation system and its many advantages over gravity flow when a single main is used have practically eliminated this system from consideration for new installations, though many systems of this type installed in years past are still functioning.

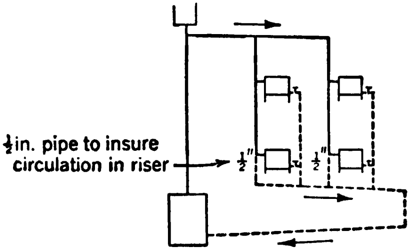
239. Two-Pipe or Two-Main Up-Feed Systems. These systems differ from the one-pipe up-feed system in that a separate return main is provided. The return water is not mixed with the supply water so that the supply water is not cooled as rapidly in the main. The general arrangement of riser connections to the radiators is the same. The supply branch connections are taken from the top of the main. Return-branch connections made at the side of the return main are preferable. The supply main is graded upward 1 in. in 10 ft in the



(a) Single-main upfeed system (b) Two-main upfeed reversed return system


Circulating pipe to prevent freezing of water in the expansion tank
Air vent for entire system

(c) Two-main upfeed direct-return system (d) Overhead down-feed system with single risers and a direct return



(e) Overhead down-feed system with double risers and a reversed-return

FIG. 185. Diagrammatic arrangements of five different hot-water heating systems.

direction of flow from the boiler, and the return main is given the same grade downward toward the boiler.

The installation as shown by Fig. 185*b* has a reversed-return main. The friction losses through the various circuits are more nearly the same with the reversed return than with a direct return, as shown by Fig. 185*c*. With the reversed-return main all radiators tend to heat simultaneously, whereas with the direct return those radiator groups nearest to the boiler have the most rapid circulation of water. Although the reversed-return main may sometimes require a little more pipe than the direct return, its use is justified in reducing the sluggishness of radiators at some distance from the boiler or heater.

Two-pipe systems permit the reduction of the supply-main size as the radiator branches are taken off. The size of the return main increases as it approaches the heater and should equal, at the heater inlets, the size of the supply main leaving the heater outlets. A manually operated air vent similar to the one shown in Fig. 186 is required at the top of each radiator.

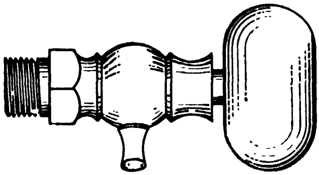


FIG. 186. Compression air cock.

240. Overhead Down-Feed Systems. These systems have an attic supply main and a basement return main. The heated water is carried directly from the heater by means of a main riser to the attic or overhead pipe space as illustrated by Figs. 185*d* and 185*e*. The main riser may be one pipe size smaller than the supply main leading to and away from it and should be adequately supported at its base. The upper end of the main riser should be enlarged by the use of a tee at least two pipe sizes larger than the main riser. The chamber thus formed aids in the elimination of air which is discharged from all parts of the system at this point through the expansion-tank vent. A properly installed overhead down-feed system requires no air valves at the radiators.

The attic supply main is taken from the side outlet of the tee at the top of the main riser and is graded downward in the direction of water flow 1 in. per 10 ft. The attic main decreases in size as the branch connections to the separate risers are taken off. All branch connections to down risers are taken from either the side or the bottom of the attic supply main. These branch connections must *never* be taken from the top of the attic supply main as they will become air bound and render the system inoperative.

The radiators may be served by single down risers which carry both supply and return water as shown in Fig. 185*d*, or they may have separate supply and return risers as in Fig. 185*e*. In any overhead down-feed hot-water system the supply connection should always be at the top of the radiator with the control valve at the outlet tapping at the bottom of the radiator. The single risers have the disadvantage that the top radiator receives the hottest water and the lower radiators receive cooler water. The separate supply and return risers allow the water to be delivered at a more nearly uniform temperature at the radiators. The radiator outlet connection with the control valve can be placed at either end of the radiator. When double risers are used the supply riser should always be by-passed through either a ½- or a ¾-in. pipe connection to the return main. This connection maintains hot water throughout a supply riser when the lowest radiator is shut off.

The return main increases in size as the return branches are attached to it and grades down toward the heater 1 in. in 10 ft. A direct or a reversed return may be used, the latter being preferable as more uniform heating by the system is secured. An exception to this recommendation is in a down-feed system in which the boiler is placed on the same level as the radiators. In this special case it has been found that the system operates more satisfactorily if a direct return is used.

241. Hot-Water-Heating-System Piping Details. Air will collect at all high points in the piping and radiators unless vented. Care must always be taken to permit the removal of air from all parts of the system where water flow occurs in order that the system or parts of it will not be inoperative. All piping should have uniform grades. Mains grading downward in the direction of flow cannot be relayed or lifted in elevation unless an air vent is provided at the top of the relay. Long-sweep elbows should be used wherever possible, and the pipes should be free from internal burrs wherever they have been cut and threaded. Provision must be made by properly installing and grading the pipes for complete drainage of the system at the boiler or heater. Wherever reductions in pipe sizes are made in horizontal lines eccentric reducing fittings or couplings are necessary, and these must be installed so that the tops of all sections of pipe joined are in alignment thereby preventing air pockets. When two or more mains branch from a common header at the boiler or heater all must be arranged so that their tops are at a common level.

242. Expansion Tanks. All systems require some arrangement to take care of the increase of water volume when it is heated; otherwise excessive and dangerous pressures will result at times. Open systems universally employ an expansion tank to absorb the increase of volume when the water is heated. When an open-tank system is cold the water level in it should be near the bottom of the tank located with its lower part at least 3 ft above the top of the highest radiator.

The volume of the expansion tank required depends upon the amount of water in the system and its temperature range together with the type of system, i.e., open or closed. The maximum range of water temperature to be considered in an open-tank system is from 35 to 210 F. The densities of water at these two temperatures are 62.42 and 59.88 lb per cu ft, respectively. Hence, when 1 cu ft of water is heated from 35 to 210 F, its final volume is 62.42 ÷ 59.88 = 1.0425 times the original volume, which is an increase of 4.25 per cent. On this basis, the volume of the expansion tank should be 4.25 per cent of the volume of the system. Extra capacity is necessary in the expansion tank to take care of abnormal conditions and possible surges of the water. Excess capacity percentages based on amounts of direct radiation of open-tank systems are: less than 3000 sq ft, 100 per cent; and above 3000 sq ft, 70 per cent.

The water volume of systems is based on the internal capacity of their radiators, as determined by Table 58, Chap. 7. In order to account for the volume of the piping and the heater, the volume of the system is taken as twice that of column radiators and 2.75 times that of tubular radiators.

Dimensions and capacities of commercial tanks suitable for storing water-volume increases of open-tank systems are listed in Table 81.

TABLE 81

EXPANSION TANKS

Capacity, Gal	Size, In.	Capacity, Sq Ft of Radiation	Capacity, Gal	Size, In.	Capacity, Sq Ft of Radiation
8	10 × 20	250	26	16 × 30	950
10	12 × 20	300	32	16 × 36	1300
15	12 × 30	500	42	16 × 48	2000
20	14 × 30	700			

The dimensions of larger tanks than those listed are ascertained by calculations based on the necessary capacity. The tank diameters vary by 2 in. and the lengths by 6 in. A convenient ratio of length to diameter is 3 to 1.

The location and connections of open expansion tanks are the best when they tend to facilitate the removal of air from the system. Expansion tank connections are included in Fig. 185. Discharge of overflow water through air vents extending above the roof is not desirable. The most satisfactory method of caring for an overflow is to return a pipe, attached to the vent above the tank, to a drain near the boiler in the basement. Vent pipes less than 1 in. in diameter are not satisfactory, and in cold localities outside vents require 1¼- to 1½-in. pipe on account of the danger of their closure by frost accumulations. Expansion tanks located in cold attics can be protected in up-feed systems by a circulating pipe as indicated in Fig. 185*c*.

Overhead down-feed systems do not require a circulating pipe if the expansion tank can be located reasonably close to the main riser. A water-gage glass attached to the tank tappings for the purpose of indicating the water level is very desirable but is not absolutely necessary.

243. Open and Closed Expansion-Tank Hot-Water Systems. All the hot-water systems thus far described are illustrated as open systems, but they may be converted to or installed as closed systems. The water should never boil in any part of a hot-water heating plant. Open systems are limited in the matter of water pressures and the corresponding boiling temperature of the water.

Pressure-systems or closed-tank systems carry hot water at higher temperatures than open systems without danger of boiling. Their pressures are limited by the safe working pressure of the structurally weakest portion of the equipment. As a result of being able to use water of high temperature, the pipes and radiators may be smaller than those necessary with open-tank systems.

244. Pressure Systems. The water pressure at any point in an open-tank system varies directly as the distance of the point below the level of the water in the expansion tank. The temperature at which the water will boil at any location is dependent upon the absolute pressure exerted upon it, as given by Table 4, Chap. 1.

Additional water pressure may be developed in an up-feed hot-water heating system by installing a closed expansion tank in the basement in the manner of Fig. 187. The tank is only partially filled with water as some space is required for air which is compressed as expansion of the water in the system forces water into the tank and the pressure increase occurs. A pressure-relief valve is attached to the return piping in order to avoid excessive or dangerous pressures. When the pressure in the system exceeds the limit allowable the relief valve opens for water discharge until the pressure drops to that for which it is set. The relief valve should not function frequently as fresh water is required after the relief valve has discharged water. Repeated feeding of raw cold water containing mineral and other materials will lead to the deposition of scale in the heater.

If a closed expansion tank is properly sized for the system it is to serve there will be no loss of water from the relief valve as long as the expansion tank contains the necessary amount of air. Heating and cooling of the system will gradually cause a transfer of air from the expansion tank to the tops of the radiators so that occasional draining of the tank is necessary to restore the necessary volume of air. Discharge of water from the relief valve is an indication that the expansion tank does not contain a sufficient amount of air.

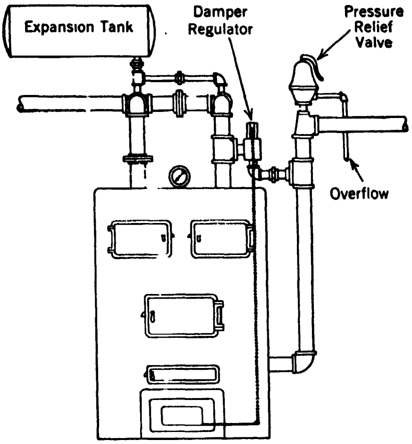
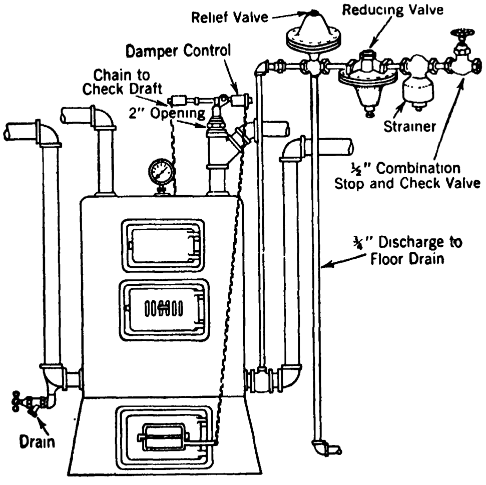


FIG. 187. Closed-tank pressure hot-water heating system.

245. Pressure Systems without Expansion Tanks. These systems are equipped with pressure-relief valves and automatic water feeders and may be used when good make-up water is available. The details of a Mueller system are given by Fig. 188. When the pressure within the system becomes excessive or greater than that for which the relief valve is set the relief valve discharges water. When the excess pressure is relieved



(*Mueller Co.*)

FIG. 188. Pressure hot-water heating system without expansion tank.

the water feeder or pressure-reducing valve allows water from a service line to enter and replace that lost.

Although there are many systems of this type in operation they are not recommended because frequent addition of water to the system is inevitable when no expansion tank is provided. The reduction in the first cost of the system through elimination of the expansion tank is usually a very small percentage of the total cost.

246. The Sizing of Closed Expansion Tanks. The function of a closed expansion tank in a hot-water system is to maintain sufficient pressure to keep the highest radiators filled when the system is cold and prevent the creation of dangerously high pressures when the system is hot. This regulation of the pressure in a closed system is provided through the expansion and contraction of an adequate volume of air which is trapped in the upper portion of the expansion tank. The greater the volume of air trapped in the tank the less will be the variation in pressure as the temperature of the water in the system fluctuates through its operating range. The procedure to be used in calculating the required size of tank for any given system can best be illustrated by an example.

Example. Calculate the size of a closed expansion tank for a system having a volume of 200 gal. The highest radiators in the system are located 30 ft above the initial water level in the tank which is to have a volume such that the pressure in the tank will not exceed 25 psig when the average water temperature in the system is 200 F. Assume that the system will be filled with water at a temperature of 60 F and that the barometric pressure is 14.7 psi.

Solution. Three different pressures and three corresponding volumes must be considered. Before the system is filled with water the tank will be completely filled with air, and the pressure will be atmospheric. Therefore, $P_1 = 14.7$ psia and $V_1 =$ the required volume of the tank. When the system is filled with cold water, a portion of it will enter the lower part of the expansion tank, compressing the air in the tank until the pressure is equal to the hydrostatic head due to the column of water extending to the top of the highest radiators. Therefore $P_2 = 14.7 + (30 \times 62.34) \div 144 = 14.7 + 13 = 27.7$ psia, and V_2 is the volume of the air after it has been thus compressed. When the average temperature of the water in the system is then raised to 200 F a volume of water equal to the expansion for the entire system is forced into the expansion tank further compressing the air and raising the pressure to a maximum. Therefore $P_3 = 14.7 + 25 = 39.7$ psia and V_3 is the volume of the air when subjected to this pressure.

Since the water of the system does not circulate through the expansion tank the temperature of the air in the tank may be assumed to remain constant, in which case $P_3V_3 = P_2V_2$. $V_3 - V_2 =$ the expansion in the system which is $200(62.34 \div 60.13) - 200 = 207.3 - 200 = 7.3$ gal or 0.977 cu ft. Substituting $V_2 - 0.977$ for V_3 in the above equation $39.7\ (V_2 - 0.977) = 27.7V_2$ and solving, V_2 is found to be 3.23 cu ft. Likewise $P_2V_2 = P_1V_1$ or $27.7 \times 3.23 = 14.7 \times V_1$, and $V_1 = (27.7 \times 3.88)/14.7 = 6.09$ cu ft. Since one cubic foot = 7.48 gal the volume of the tank in gallons = $6.09 \times 7.48 = 45.5$ gal. This is the minimum volume of the closed expansion tank for this system and does not allow for any loss of air from the tank, nor does it allow for a pressure greater than atmospheric at the top of the radiators. The volume of the installed tank should be approximately double the minimum volume or around 90 gal.

247. Computed Weights and Volumes of Water Handled per Hour. Expressions for the weight and volume of water to be handled per hour by radiators are easily stated in equation form.

Weight of water,

$$H_r = W_w(t_1 - t_2)$$

$$W_w = \frac{H_r}{(t_1 - t_2)} \tag{105}$$

Volume of water in cubic feet,

$$H_r = Q \times d_w(t_1 - t_2)$$

$$Q = \frac{H_r}{d_w(t_1 - t_2)} \tag{106}$$

Volume of water gallons,

$$H_r = \frac{G \times d_w \times (t_1 - t_2)231}{1728}$$

$$G = \frac{H_r \times 1728}{231 \times d_w(t_1 - t_2)} \tag{107}$$

where H_r = heat given off by the radiators, Btu per hr.
W_w = weight of water handled, lb per hr.
t_1 = inlet water temperatures, deg F.
t_2 = outlet water temperature, deg F.
Q = volume of water handled per hour, cu ft.
d_w = mean density of water handled in the radiators, lb per cu ft.
G = gallons of water circulated per hour.
1728 ÷ 231 = gal per cu ft

Example. A tubular radiator rated at 100 sq ft of equivalent direct surface stands in air maintained at 65 F when t_1 is 180 F and t_2 is 160 F. The supply and return connections are $1\frac{1}{4}$-in. pipe. Find the weight and volume of water handled per hour; also the average velocity of flow in the supply and return pipes.

Solution.

$$R' = 150\left[\frac{170 - 65}{170 - 70}\right]^{1.3} = 159.8 \text{ Btu per hr per sq ft}$$

$$H_r = 100 \times 159.8 = 15{,}980 \text{ Btu per hr}$$

$$W_w = 15{,}980 \div (180 - 160) = 799 \text{ lb per hr}$$

$$d_w = 60.8 \text{ lb per cu ft at } 170 \text{ F}$$

$$Q = 15{,}980 \div 60.8(180 - 160) = 13.1 \text{ cu ft per hr}$$

$$G = (13.1 \times 1728) \div 231 = 98 \text{ gal per hr}$$

The average velocity of flow v in the supply and return pipes is calculated as

$$v = \frac{Q}{60 \times 60 \times A} = \text{fps}$$

where A = internal cross-sectional area of pipe, sq ft.

$$v = \frac{13.1 \times 144}{60 \times 60 \times 1.496} = 0.348 \text{ fps.}$$

Practical values of actual velocities of water flow in gravity systems when the water operating temperatures are 180 and 160 F range from 0.25 to 1 fps for pipe sizes from 1 to 6 in., and from 0.5 to 2 fps for pipes larger than 6 in. in diameter,

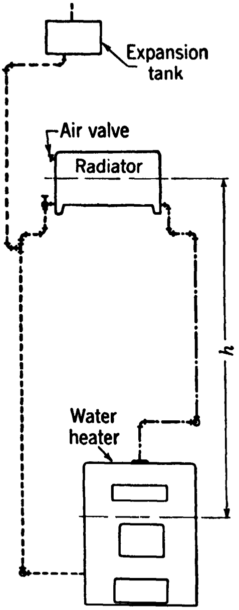


Fig. 189. Elementary gravity-flow hot-water heating system.

248. Flow Production in Gravity Systems. The motive head which produces flow is due to the difference in weights of two columns of water of equal height and equal cross section. These column heights h are indicated by the sketch, Fig. 189, of an elementary hot-water system having a single radiator. The conditions are analogous to those of the chimney discussed in Art. 117, except that water is the fluid instead of flue gases. The heights h of the columns are taken as the distance between the mean elevation of the water space in the radiator and the mean elevation of the water space in the heater.

The pressure difference P is $P = h(d_{w2} - d_{w1})A$ lb per sq ft when the water densities in pounds per cubic foot are d_{w2} for the cool side and d_{w1} for the warm side and A, the cross-sectional area of the columns, is one square foot. It is desirable to express the motive head either in feet of water at the mean density of the water flowing or in terms of milinches. The later is $\frac{1}{1000}$ in., or 1 in. is equal to 1000 milinches. The mean density of the water flowing is $(d_{w2} + d_{w1})/2$, and the motive head h' ft to produce flow is $2P/(d_{w2} + d_{w1})$, or

$$h' = \frac{2h(d_{w2} - d_{w1})}{(d_{w2} + d_{w1})} \tag{108}$$

In terms of milinches, equation 108 becomes

$$h_{mi}' = \frac{1000 \times 12 \times 2h(d_{w2} - d_{w1})}{(d_{w2} + d_{w1})} \tag{109}$$

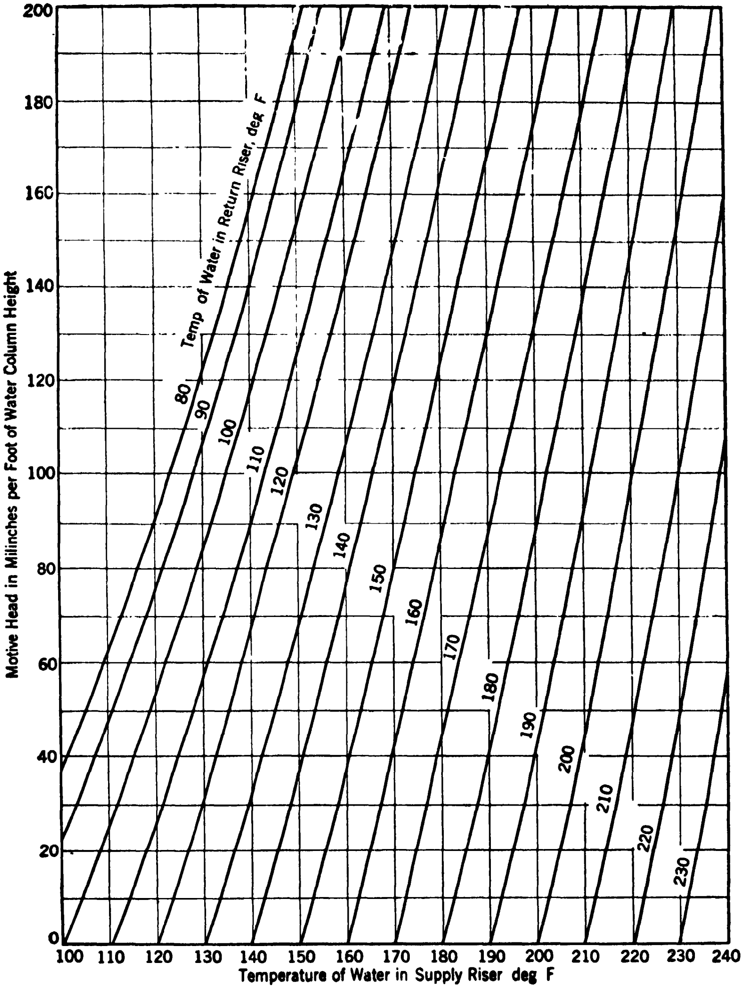


FIG. 190. Head available per foot of water column in gravity-flow hot-water heating systems.

Data for the available head in milinches per foot of mean height are given for several inlet and outlet water temperatures at radiators by Fig. 190. As a matter of interest the theoretical velocity of flow corresponding to the available head h' can be ascertained. This velocity is

$$v = \sqrt{2gh'} \text{ fps} \tag{110}$$

Example. The system of Fig. 189 operates with water in the supply riser at 190 F. ($d_{w1} = 60.35$) and in the return riser at 170 F ($d_{w2} = 60.79$). The mean height of the system, h, is equal to 10 ft. Find the available head in feet and milinches and the theoretical velocity of flow.

Solution. Available head in feet, $h' = 2 \times 10(60.79 - 60.35)/(60.79 + 60.35) = 0.073$. Available head in milinches, $h_{mi}' = 1000 \times 12 \times 0.073 = 875$. Theoretical velocity, feet per second, $= v = \sqrt{2 \times 32.16 \times 0.073} = 2.16$.

249. Mechanical-Circulation Systems. Forced-circulation systems differ from those of gravity flow in that a pump or booster is installed in the return piping near the boiler. The function of the pump is to create mechanically a sufficient head to overcome the resistances offered to flow in the circuits of a hot-water heating plant. The pumps may be either very small units installed to improve the operation of gravity systems, Fig. 191, or units of a size and strength to circulate water under considerable pressure as in Fig. 192. In large forced-circulation systems the expansion-tank connection should be in the suction line near the pump as indicated by Fig. 192.

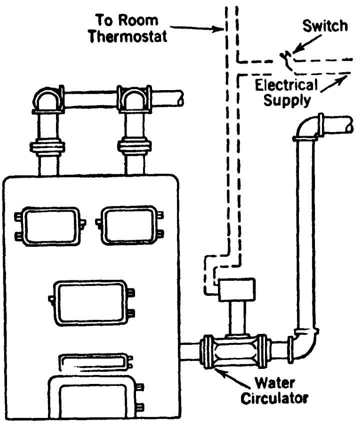


FIG. 191. Hot-water heating system with water circulator.

Mechanical circulation insures positive flow of the water irrespective of the mean height of the system or the drop in water temperature. As a result of the greater velocities of flow with forced circulation than are obtainable with gravity-flow the sizes of the system pipes may be made smaller. The gravity head produced in a forced circulation is so small in comparison with the head created by the pump that it usually may be ignored.

Forced-circulation hot-water plants are suitable for tall buildings if the building is divided into zones with the heaters and pumps located in the different zones. Equipment so placed is relieved of the major portion of pressure to which it would be subjected if placed in the basement. Forced-circulation hot-water systems have applications in buildings covering considerable ground areas and are employed to a limited extent in district heating. For underground hot-water mains in tunnels and conduits much of the discussion of Art. 234, is applicable.

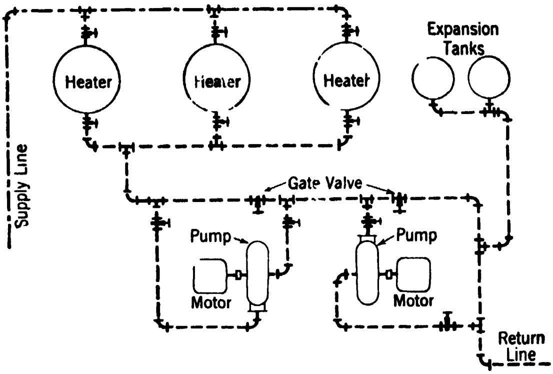
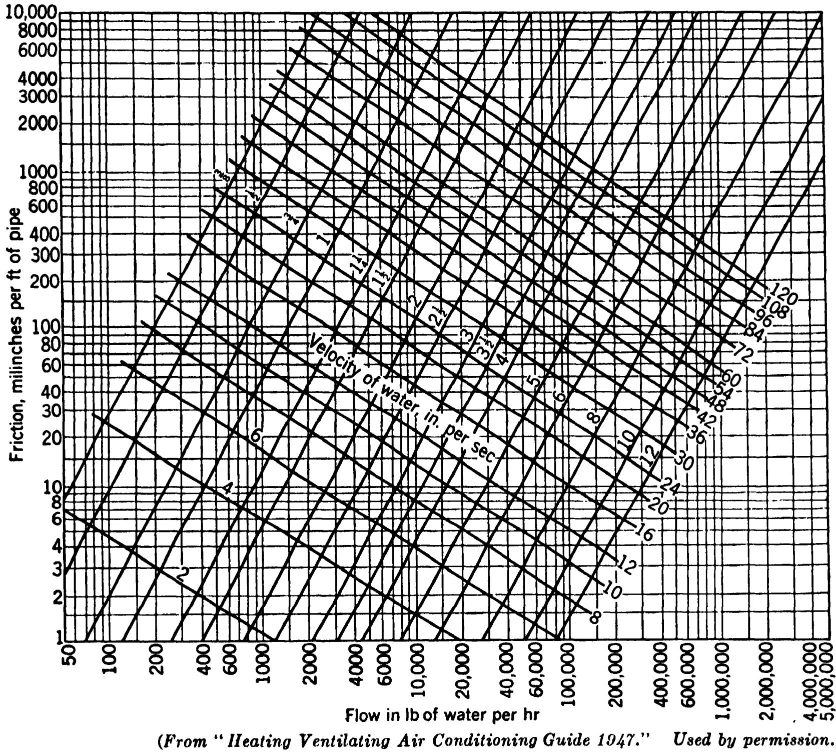


FIG. 192. Pump and heater arrangement for forced-circulation hot-water heating system.

250. Resistances to Flow in Hot-Water Heating Systems. The available head in gravity-flow systems is used entirely to overcome friction and other losses in the circuit traveled when water flow has been established.

The available operating head is consumed by: entrance losses as the water passes from the heater to the supply main; entrance losses as the water leaving radiator enters the return piping; friction in the valves, fittings, and pipes; and turbulence and internal friction in the water. Account must be taken of the various losses when the sizing of the various pipes is based on the resistances of the circuit of which they are part.

The experiments and research by Dr. F. E. Giesecke have established that the friction losses in clean commercial pipes vary as the 1.78 power of the velocity of flow, in feet per second, and in fittings as about the square of the velocity of flow. Friction losses are also dependent upon the water density, the size of pipe, and the characteristics of the internal surfaces of the pipe. For new, clean, commercial steel and wrought-iron pipes, the friction losses, in terms of milinches of available head used up per foot of pipe, as determined by Dr. F. E. Giesecke, are given by Fig. 193. These losses per foot of pipe are given for several sizes of pipe with different rates of water flow in pounds per hour. As determined the losses are for a system with water at 190 F in the flow risers and 170 F in the returns. Although the friction losses vary somewhat with change of water density and its absolute viscosity the data of Fig. 193 may be used with small error for other temperature differentials and water densities, provided that the rates of flow are the same in a given size of pipe.



(*From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.*)

FIG. 193. Friction head per foot of black-iron pipe in hot-water heating systems.

As the losses produced by fittings vary almost as the square of the velocity of flow their frictional equivalents can be expressed with small errors in terms of one fitting such as a 90-deg elbow. Elbow equivalents for iron and copper fittings are given in Table 82.

The elbow equivalent of tees is greatly affected by the condition of flow. The resistance of this fitting in a supply main may be neglected

TABLE 82

IRON AND COPPER ELBOW EQUIVALENTS*

Fitting	Iron Pipe	Copper Tubing	Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0	Open-globe valve	12.0	17.0
Elbow, 45-deg	0.7	0.7	Angle radiator valve	2.0	3.0
Elbow, 90-deg long-turn	0.5	0.5	Radiator or convector	3.0	4.0
Elbow, welded 90-deg	0.5	0.5	Boiler or heater	3.0	3.0
Reduced coupling	0.4	0.4	Tee, when used in place of an elbow	1.8	1.2
Open return bend	1.0	1.0			
Open gate valve	0.5	0.7			

* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

as far as the main circuit is concerned. The following equation[1] may be used to determine the elbow equivalents of tees used in a supply main or in a return main when estimating the resistance in a branch circuit where a portion of the main stream is diverted through the side outlet of the tee or where flow in a branch is returned to a main through a similar arrangement.

$$E = 0.75\left(\frac{v_m^2 + v_b^2}{v_b^2}\right) \tag{111}$$

where E = elbow equivalent of the tee.

v_m = velocity of the combined stream, in. per sec.

v_b = velocity in the branch, in. per sec.

v_m and v_b may be obtained from Fig. 193 if iron pipe is used or from Fig. 194 when the flow is through copper tubing. For a tee in a return main the stream of water entering from a branch disturbs the flow of the stream passing straight through, and the friction loss caused by this disturbance should not be neglected. The elbow equivalent of such an arrangement to be included in the resistance of the return main is given by the following:

$$E = 0.50\left(\frac{v_d^2 + v_u^2}{v_u^2}\right) \tag{112}$$

where E = elbow equivalent of the tee.

v_d = velocity downstream from tee, in. per sec.

v_u = velocity upstream from tee, in. per sec.

---

[1] "The Loss of Head in Cast Iron Tees, by F. E. Giesecke, W. H. Badgett and J. R. Eddy, *Bulletin* 41, Texas Engineering Experiment Station, College Station, Texas.

Fittings at lettered points (see Fig. 196) in the supply main should be included in the downstream section. Fittings at lettered points in the return main should be included in the upstream section.

After the sum of the elbow equivalents of fittings has been found for the portion of the circuit which is under consideration it may be converted to equivalent length of straight pipe or tubing in feet by

$$L = \left(\frac{25 \times D \times E}{12}\right) \tag{113}$$

where D = the nominal diameter of the pipe or tubing, in.
E = the sum of the elbow equivalents.

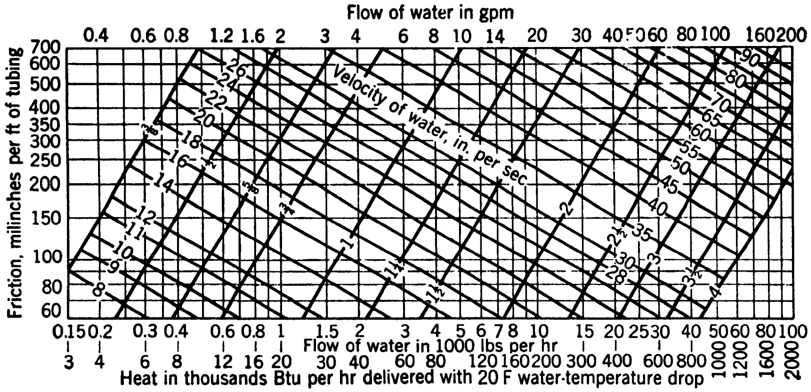


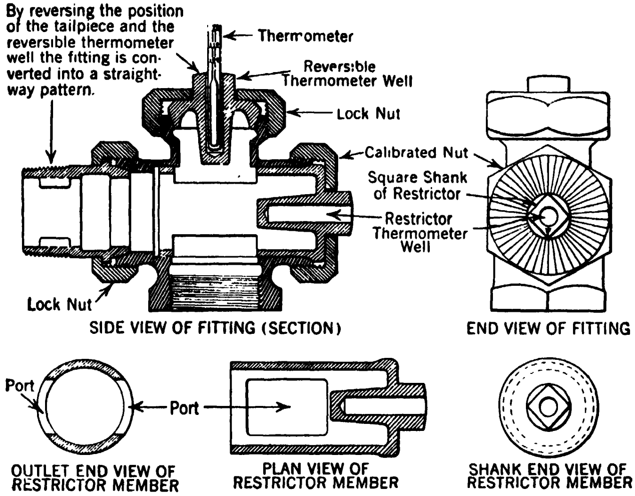
FIG. 194. Friction head of water flowing in Type-L copper tubing.

The total equivalent length of the section being considered is then found by adding the equivalent length of the fittings and the actual length of the pipe or tubing.

The friction loss in a circuit constructed of iron pipe and fittings is given per foot of equivalent length by the chart of Fig. 193. Figure 194 includes similar data which are applicable to piping systems in which copper tubes and fittings have been used.

251. Systems as Affected by Pipe Sizes. The desirable condition is that the friction losses in all pipe circuits be the same and just equal to the total head available. This condition is practically impossible with commercial pipe and fittings. At best the design can usually only approximate the desired conditions, and compensations must be made for inexactness.

In gravity systems the effects of all pipes too small are to retard the flow of water and increase its temperature drop which in turn tends to increase the operating head but also reduces the amount of heat given off by the radiators. The effect of pipes too large are: reduced temperature drop of the water passing through the system, increased flow of water and heat transmission of radiators, and reduced operating head. Variable resistances in the circuits caused by improper pipe sizes, pipe burrs, and obstructions produce non-uniform heating of the system. Gravity systems are self-compensating in many respects so that systems laid out with inexact pipe sizes often, but not always, adjust themselves to a fairly satisfactory operating condition. Where the system cannot bring itself into sufficient adjustment, placing either variable orifice valves, adjustable restrictor fittings, Fig. 195, in the radiator circuits or thin copper orifice plates in the unions of radiator connections helps to compensate for oversized pipes. Where the pipe sizes are all too small, in a gravity system, changes of operation may be effected by means of a circulator. Forced-circulation systems have the motive force for the water created at one place, the pump. The selected pipe sizes are those of the minimum size and cost of installation which will not unduly increase the required head to be maintained by the pump and the expense of operation.


By reversing the position of the tailpiece and the reversible thermometer well the fitting is converted into a straightway pattern.
Thermometer
Reversible Thermometer Well
Lock Nut
Calibrated Nut
Square Shank of Restrictor
Restrictor Thermometer Well
Lock Nut
Port
Port

SIDE VIEW OF FITTING (SECTION)

END VIEW OF FITTING

OUTLET END VIEW OF RESTRICTOR MEMBER

PLAN VIEW OF RESTRICTOR MEMBER

SHANK END VIEW OF RESTRICTOR MEMBER

Fig. 195. Oriflow restriction fitting.

The water may flow through a forced-circulation system so that it will cool from 10 to 30 deg F or any other reasonable number of degrees in the radiators. Generally a water temperature drop of 20 deg F in the system gives good results without excessive power requirements for circulating the water. Such operation of a system requires pumping only half the amount of water necessary when the decrease in water temperature is limited to 10 deg F and allows for a future expansion of 50 per cent in capacity without increase in the amount of water handled per hour before exceeding the maximum drop of 30 deg F in the water temperature.

Heat losses from insulated mains and pipes can usually be ignored. However, if it seems desirable, the actual amount of surface in the radiators can be decreased to compensate for the heat given off by the mains, etc., after the pipe sizes have been determined. When forced-circulation systems are used in residences, the mains and all other pipes are so small that the heat loss from them is negligible, even when they are not insulated.

252. Sizing of Pipes for Hot-Water Heating Systems. In general the procedures for sizing the pipes of gravity-flow and forced-circulation hot-water heating systems are much the same. With gravity-flow systems the available head in a radiator circuit is dependent on the heights of two water columns and their differences in density. The design conditions thereby limit the available head in gravity-flow installations, and great care must be exercised in order that the friction losses for the conditions of the design do not exceed the head available in any water circuit. If a pipe is sized too small it cannot carry the requisite amount of water to supply the necessary heat to a radiator operating on the basis of a fixed drop in the temperature of the water within it with a given amount of available head. With forced circulation the available head is not limited by differences in water temperatures and elevation, and the problem becomes one of selecting pipes of such sizes that the required amount of water can be handled with a reasonable expenditure for power to drive the pump and an economical first cost for the installed plant.

In either system the design should be made so that the resistances to flow in all water circuits are such that some circuits are not favored over others, thereby causing one portion of the building to heat more readily than other portions. With commercial pipes of uniform sizes in the different sections it is practically impossible to balance the circuits so that all function to give exactly the desired results. Therefore it may become necessary to introduce additional resistance in some water circuits to balance their friction losses with those of other circuits. This can be done by inserting in those circuits where additional resistance is required either sections of pipe of a smaller size, balancing elbows and valves, or orifices which create an obstruction to flow. Table 83, based on the work of Dr. Giesecke, gives frictional data for central-opening circular-diaphragm orifices when placed in commercial pipes

TABLE 83

FRICTION HEADS (IN MILINCHES) OF CENTRAL CIRCULAR-DIAPHRAGM ORIFICES IN UNIONS*

Diameter of Orifices, In.	Velocity of Water in Pipe, In. per Sec									
	2	3	4	6	8	10	12	18	24	36
¾-in. Pipe										
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5,700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2,900	5,200	8,000	12,000	26,000	47,000	
0.40	170	380	660	1,500	2,600	4,000	6,800	13,000	24,000	53,000
0.45	...	185	330	740	1,300	2,000	2,900	6,500	12,000	27,000
0.50	...	...	155	350	620	970	1,400	3,200	5,700	13,000
0.55	...	...	75	170	300	480	700	1,600	2,800	6,400
1-In. Pipe										
0.35	900	2000	3500	7,800	14,000	22,000	32,000			
0.40	460	1000	1800	4,000	7,200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2,300	4,100	6,400	9,300	21,000	37,000	
0.50	160	330	580	1,400	2,300	3,700	5,400	12,000	22,000	50,000
0.55	...	190	330	750	1,300	2,200	3,000	7,000	13,000	28,000
0.60	...	...	200	440	800	1,300	1,800	4,200	7,400	17,000
0.65	...	...	120	260	460	720	1,100	2,400	4,300	10,000
1¼-In. Pipe										
0.45	1000	2250	4000	8,900	16,000	25,000	36,000			
0.50	660	1450	2600	5,800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3,800	6,800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2,500	4,400	6,900	10,000	22,000	40,000	
0.65	190	420	750	1,700	3,000	4,700	6,700	15,000	27,000	60,000
0.70	...	285	510	1,150	2,000	3,100	4,500	10,000	18,000	40,000
0.75	...	190	330	750	1,300	2,100	3,000	6,700	12,000	26,000
1½-In. Pipe†										
0.55	850	1900	3300	7,400	13,000	21,000	30,000			
0.60	600	1300	2300	5,400	8,600	16,800	21,000	50,000		
0.65	400	850	1500	3,600	7,200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2,600	4,400	7,000	10,000	21,000	39,000	
0.75	180	400	760	1,800	3,000	5,000	7,000	14,000	28,000	
0.80	...	300	540	1,200	2,200	3,200	5,000	10,200	19,000	45,000
0.85	...	200	380	860	1,600	2,300	3,000	7,800	13,000	30,000
2-In. Pipe†										
0.70	890	1850	3500	7,400	14,000	22,300	33,000			
0.80	470	975	1800	3,900	7,400	11,700	17,000	37,000		
0.90	255	560	1000	2,200	4,200	6,500	9,500	20,500	38,000	
1.00	160	340	610	1,320	2,520	4,000	5,800	12,500	23,000	49,000
1.10	...	214	375	850	1,600	2,500	3,700	7,900	14,000	30,000
1.20	...	...	195	460	950	1,360	1,910	4,200	8,100	16,800
1.30	...	...	...	275	525	980	1,375	3,100	4,400	8,850

* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

† The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

carrying water with different velocities of flow. Two parallel circuits can easily be balanced if a suitable cock is included in each one near the point where they are joined.

In gravity flow the available head and the quantity of water to be handled fix the pipe size. With forced circulation the friction losses per foot of pipe are selected between reasonable limits and the pipes sized on the basis of the amount of water which they must handle and the allowable friction losses per foot of pipe. As the latter calculations are somewhat simpler than those for gravity flow the pipe sizing for forced-circulation systems will be considered first.

253. Piping Design for Forced-Circulation Hot-Water Systems. The methods of procedure given in this article were suggested by Mr. R. T. Kern and may be used for systems to which the simplified procedure of Art. 257 is not applicable.

As the velocity of flow is greater with forced circulation than with gravity flow, the head required to produce the flow must also be greater for a given size of pipe. There is a point where the velocity of flow and the required pressure head in a forced-circulation system results in the minimum cost of installation and operation. The economical pressure head to be created should be estimated for each particular system. Under ordinary conditions the economical friction pressure losses will amount to 140 to 400 milinches of water per ft of run, and the pressure head to be created per foot of run should fall between these limits. Friction losses less than 140 milinches per ft in forced-circulation systems require excessively large pipes. With commercial pipes it is not always possible to maintain the friction pressure losses above 140 milinches per ft. Losses of more than 400 milinches per ft are excessive and should be used with caution. Where one section of the piping is short compared to others in the same system, it may be advisable to use a pipe size such that the friction loss will be greater than 400 milinches per ft to equalize the friction losses with those of longer conduits.

The following example illustrates the procedure to be followed in designing forced-circulation hot-water heating systems.

Example for a Two-Main Reversed-Return System. Assume a system having five radiators, one boiler, and a pump as indicated by Fig. 196. The lengths of straight pipe and the elbows of both the supply and return mains are shown by the plan, together with the heat outputs of the radiators and the weights of water to be handled by the various portions of the system. No allowances will be made for the loss of water temperature in the supply mains and pipes, and the decrease in the water temperature at all radiators will be taken as 20 F. Each radiator has 3 ft of pipe in the supply connection from the main and 4 ft of pipe in the outlet connection to the return main. All radiators have 3 elbows in the inlet connection, and radiators 1, 2, 3, and 4 also have in addition a connection through a tee to the supply main. Each radiator has at its outlet an angle valve, one 90-deg elbow, and all radiators with the exception of number 1 have a connection through a tee to the return main. Determine the pipe sizes, assuming that iron pipe will be used throughout.

Solution. As a matter of convenience in tabulating the calculated values an arrangement such as Table 84 is a convenience. The headings of the table are self-explanatory.

The values which appear in the table for section *AB* in the supply main were derived as follows. The heat supplied was obtained by adding the heat delivered by each of the 5 radiators, and the water handled was obtained by dividing this sum by 20, since a temperature drop of 20 deg was assumed. The nominal pipe size was chosen with the aid of the chart of Fig. 193. A one-inch pipe was selected, as the friction loss with this size falls in the recommended range of 140 to 400 milinches per foot. The velocity was obtained from the same chart at the same time. The length of straight pipe was obtained from Fig. 196. The elbow equivalents of the fittings was computed as follows: 2 elbows between the pump and the boiler, +3 which is the elbow equivalent of the boiler +3 elbows between the boiler and point *B*. The elbow equivalents of the fittings in section *AB* is therefore 2 + 3 + 3 = 8. The equivalent length of the fittings in this section was calculated with equation 113 as $L = 25 \times 1 \times 8 \div 12 = 16.7$ ft. The total equivalent length of the straight pipe and the fittings = 30 + 16.7 = 46.7 ft. The friction

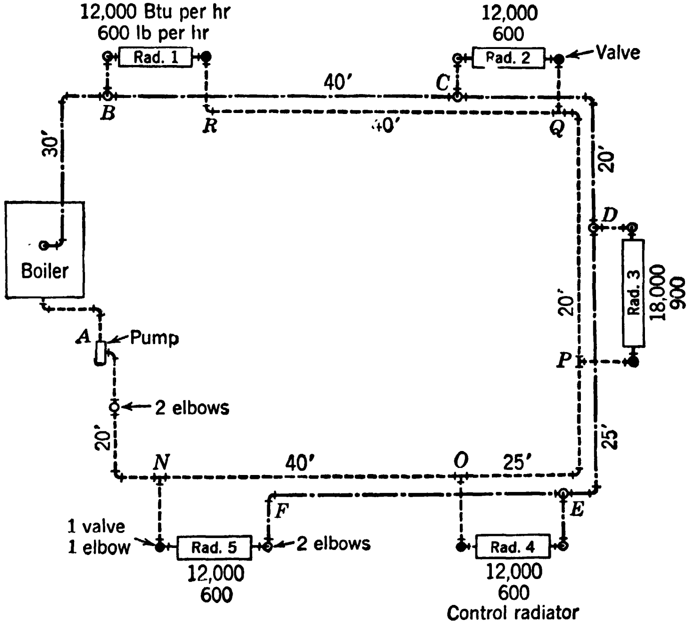


Fig. 196. Forced-circulation reversed-return hot-water heating system. Expansion-tank connections not shown.

TABLE 84

Pipe Sizing Data for Reversed-Return Forced-Circulation Hot-Water Heating System

Section	Heat Supplied, Btu per Hr	Water Handled, Lb per Hr	Pipe Size, In.	Velocity, In. per Sec	Length of Straight Pipe, Ft	Elbow Equivalents of Fittings	Equivalent Length of Fittings	Total Equivalent Length of Section, Ft	Friction Loss per Foot, Milinches	Friction Losses in Section, Milinches	Accumulative Total Friction Losses, Milinches
Supply Main											
AB	66,000	3,300	1	30	30	8	16.7	46.7	400	18,700	18,700
BC	54,000	2,700	1	24	40	0	0	40.0	280	11,200	29,900
CD	42,000	2,100	1	18	20	1	2.1	22.1	190	4,200	34,120
DE	24,000	1,200	$\frac{3}{4}$	16	25	1	1.6	26.6	180	4,790	38,910
EF	12,000	600	$\frac{1}{2}$	16	40	0	0	40.0	270	10,800	49,710
Return Main											
AN		3,300	1	30	20	4	8.3	28.3	400	11,300	11,300
NO		2,700	1	24	40	1.3	2.7	42.7	280	11,950	23,250
OP		2,100	1	18	25	2.4	5.0	30.0	190	5,700	28,950
PQ		1,200	$\frac{3}{4}$	16	20	2.1	3.3	23.3	180	4,180	33,130
QR		600	$\frac{1}{2}$	16	40	1.0	1.0	41.0	270	11,080	44,210
Radiator 1	12,000	600	$\frac{1}{2}$	16	7	13.3	13.8	20.8	270	5,600	
Radiator 2	12,000	600	$\frac{1}{2}$	16	7	12.9	13.4	20.4	270	5,510	
Radiator 3	18,000	900	$\frac{3}{4}$	13	7	13.6	21.2	28.2	125	3,660	
Radiator 4	12,000	600	$\frac{1}{2}$	16	7	12.9	13.4	20.4	270	5,510	
Radiator 5	12,000	600	$\frac{1}{2}$	16	7	12.4	12.9	19.9	270	5,380	

loss per foot (400 milinches) was obtained from Fig. 193, and the section friction loss = 46.7 × 400 = 18,700 milinches. Similar data for the other sections of the supply main were obtained in a similar manner. The resistances of the tees in the supply main were neglected in estimating the friction loss of the main.

The data applying to the different sections in the return main were computed in exactly the same manner except that, in computing the elbow equivalents of the fittings, the tees were included. The elbow equivalents of the fittings in section *NO* is the elbow equivalent of the tee at *N* which was computed by the use of equation 112 as follows:

$$E = 0.5\left(\frac{v_d^2 + v_u^2}{v_u^2}\right) = 0.5\left(\frac{30^2 + 24^2}{24^2}\right) = 0.5\left(\frac{900 + 576}{576}\right) = 1.3.$$

The data relative to each of the 5 radiator circuits were also computed in the same manner. The fittings considered in each radiator circuit were the tee in the supply main, the elbow equivalents for which were calculated by means of equation 111; 3 elbows between the supply main and the radiator; 3 elbow equivalents for the radiator; 2 elbow equivalents for the angle radiator valve, Table 82; and one elbow between the valve and the return main and the tee in the return main, the elbow equivalents for which were also calculated by means of equation 111. In using equation 111 for estimating the resistance of a tee in the return main as it affects the radiator circuit, v_b is taken as the velocity in the radiator branch circuit and v_m is the velocity in the main, downstream from the tee.

The completion of the problem involves the calculation of the friction in each complete circuit through each of the 5 radiators and the selection of the orifices or pipe changes needed for balancing the circuit. The necessary data are tabulated in Table 85.

TABLE 85

CALCULATIONS FOR BALANCING RADIATOR CIRCUITS IN THE SYSTEM OF FIG. 196

Radiator Number	Friction Loss in Supply Main, Pump to Tee, Milinches	Friction Loss in Radiator Circuit, Supply Main to Return Main, Milinches	Friction Loss in Return Main, Tee to Pump, Milinches	Total Friction Loss in Complete Circuit, Milinches	Additional Resistance Needed, Milinches	Required Alterations in Radiator Circuit
1	18,700	5600	44,210	68,510	30	None
2	29,900	5370	33,130	68,540	0	None
3	34,120	3400	28,950	66,700	1840	Include 0.5-in. orifice
4	38,910	5290	23,250	67,670	840	Change 2.0 ft of $\frac{1}{2}$-in. pipe to $\frac{3}{8}$ in.
5	49,710	5050	11,300	66,390	2120	Change 5 ft. of $\frac{1}{2}$-in. pipe to $\frac{3}{8}$ in.

The complete circuit which includes radiator 2 has the greatest friction loss, and it therefore becomes the control radiator. The difference between the friction loss through the complete circuits of radiators 1 and 2 is negligible. The radiator 3 circuit friction loss should be increased by approximately 1840 milinches. It may be noted in Table 83 that the orifice most nearly meeting the requirement is one having an opening 0.50 in. in diameter. The friction loss in the radiator 4 circuit should be increased by 840 milinches. It may be noted in Fig. 193 that the friction loss in a $\frac{3}{8}$-in. pipe handling 600 lb per hr is 700 milinches per ft., which is 700 − 270 = 430 milinches per ft. more than that for $\frac{1}{2}$-in. pipe which was assumed in making the estimate. Therefore changing 840 ÷ 430 = 1.95 ft. of the piping in the circuit of radiator 4 from $\frac{1}{2}$-in. to $\frac{3}{8}$-in. size will increase the resistance of the total circuit to approximately that of the total circuit through the control radiator. Similarly it may be found that 4.95 ft of the $\frac{1}{2}$-in. pipe in the circuit of radiator 5 should be changed to $\frac{3}{8}$ in.

The same general procedure which was used in the preceding example can be used for sizing the pipes in a forced-circulation direct-return system (see Fig. 185*c*). In a hot-water system of this type the circuits through the different radiators are of different lengths so that it is necessary to choose the radiator farthest from the boiler as the control radiator. Since this radiator is fed by the end of the supply main and since the return main originates at its outlet, the longest circuit of the system consists of the piping between the pump and the boiler, the boiler, the entire length of the supply main from the boiler to the control radiator, this radiator, and the entire length of the return main from the radiator to the pump. The friction losses in this circuit are first calculated; then the friction losses in the shorter circuits are brought up to this amount through the use of either orifices, restriction fittings, or piping of reduced size.

254. Pipe Sizing for an Up-Feed Gravity-Flow System. The sizing of the pipes may be done by the method of the following example.

Example. Use the simple gravity up-feed system of Fig. 197 which has a single radiator. The heat given off by the radiator is 15,000 Btu per hr with the water in the supply riser at 190 F and in the return riser at 170 F when the room-air temperature is 70 F. The supply riser connection has 10 ft of pipe, 5 elbows, and an angle radiator valve. The return riser includes 8 elbows and 15 ft of pipe.

Solution. The radiator must handle 15,000 ÷ 20 = 750 lb of water per hour and the available head = 7 × 87.5 (Fig. 190) = 612.5 milinches. The length of pipe in the circuit exclusive of the equivalent length of the elbow equivalents is 10 + 15 = 25 ft. The elbow equivalents of the fittings in the circuit are as follows: Each elbow 1.0, Table 82, boiler and radiator each 3.0, and angle valve 2.0. The total elbow equivalents of the fittings is computed as follows, starting with the boiler, 3 + 5 + 2 + 3 + 8 = 21. The equivalent length of the elbow equivalents cannot be calculated until a pipe size has been temporarily assumed. Because of the friction loss in the fittings, that in the straight pipe must be considerably less than 612.5 ÷ 25 = 24.5 milinches per ft. Reference to Fig. 193 reveals that the friction in a one-inch pipe transporting 750 lb of water per hr is 26 milinches per ft, which is too high; therefore $1\frac{1}{4}$-in. pipe for which the loss is 7 milinches per ft will be selected as the trial size. The equivalent length of the elbow equivalents is then $L = (25 \times 1.25 \times 21) \div 12 = 54.6$ ft, equation 113. The total equivalent length of the circuit is $25 + 54.6 = 79.6$ ft, and the friction loss of the circuit is found to be $79.6 \times 7 = 557$ milinches. The excess of available head is $612.5 - 557 = 55.5$ milinches. For a system with a single radiator the plant would adjust itself to give off very nearly the correct amount of heat. If, however, it is necessary to balance the friction losses against the available head it can be done with either an orifice in the supply riser or by reducing in size a portion of the supply riser and the included fittings. The frictional resistance of the system would be increased $26 - 7 = 19$ milinches for each foot of riser changed from $1\frac{1}{4}$-in. pipe to the 1-in. size. To exactly balance the available head with friction loss, $55.5 \div 19 = 2.9$ equivalent ft of pipe should be reduced in size.

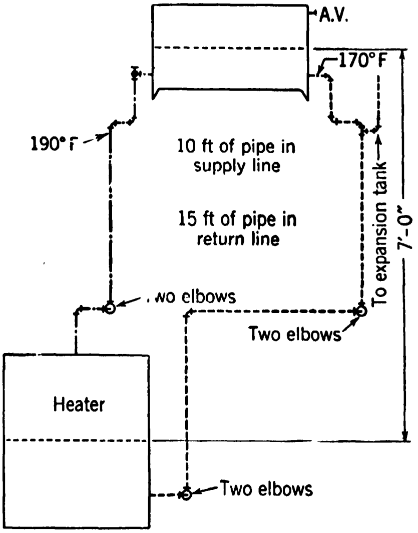
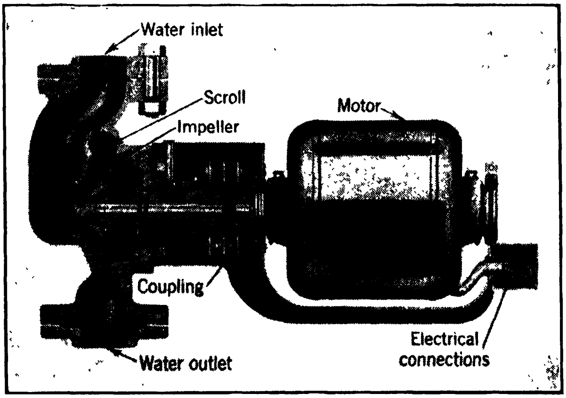


FIG. 197. Simple up-feed gravity-flow hot-water heating system.

255. Circulator Selection. Fig. 198 shows a sectional view of a small single-stage centrifugal pump suitable for service as a circulator in a hot-water heating system. Fig. 199 shows performance curves for one-manufacturer's line of circulators. Preliminary steps in the



(Trane Co.)

FIG. 198. Water circulator.

selection of the proper size are the calculation of the amount of water which must be circulated (see Art. 247) and the estimation of the friction head against which the unit must operate (see Art. 250). In general the sizing of pipes throughout the system should be such that the total head required to overcome friction is between 8 and 11 ft of water, except for small single-main systems for residences, which are discussed in Art. 257. When the required volume in gallons per minute and the total head in feet of water are known, the unit in a particular manufacturer's line of circulators, which is best suited to the system, may be selected by referring to performance curves such as those appearing in Fig. 199. The circulator selected should be the one whose curve is nearest the intersection of a vertical line representing the required gallons per minute and a horizontal line representing the total friction head of the system. If the choice is in doubt due to the point of intersection falling between the curves for two different pumps, the larger of the two pumps should be chosen.

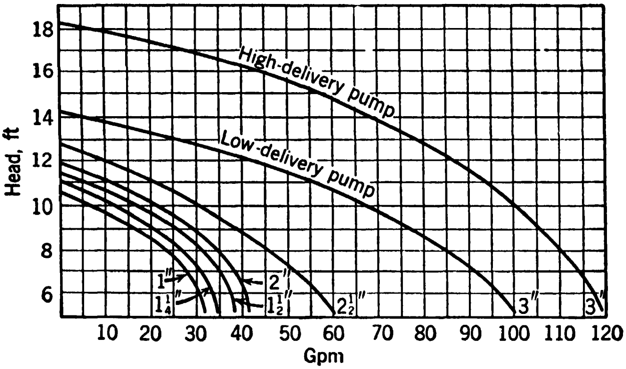


FIG. 199. Performance curves of Trane water circulators with various operating heads.

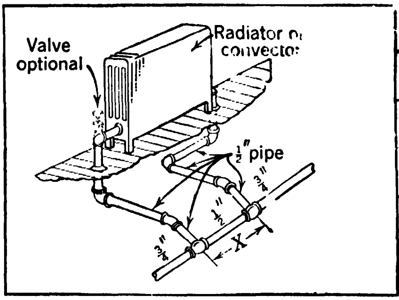
256. Power for Pumping. It is necessary to estimate power required to circulate the water through a hot-water heating system in order that the proper size of the driving motor may be specified when it is not furnished by the pump manufacturer. The power requirement in terms of horsepower is given by the following equation:

$$\text{hp} = \frac{PQ}{33000 \times e} \tag{114}$$

where P = pressure head, lb per sq ft.
Q = volume of water pumped, cfm.
e = efficiency of pump expressed decimally.

The efficiency of the pump will ordinarily be between 60 and 70 per cent.

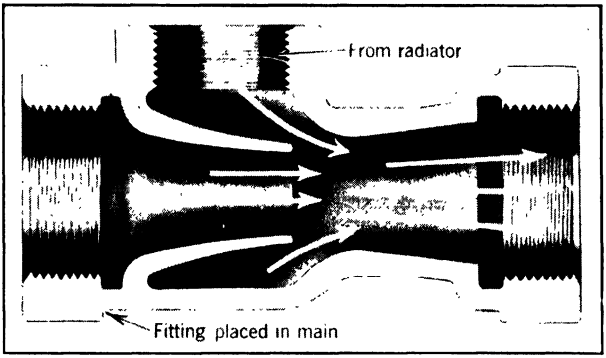
257. Design of Small One-Pipe Forced-Circulation Systems. The simple method which follows is that outlined in the *I = B = R Installation Guide*[2] for residences and small buildings which have heat losses less than 60,000 Btu per hr. Figure 200 indicates the arrangement of the pipe and fittings connecting a radiator to a main as shown by the Guide, which specifies standard pipe and screwed fittings in all parts of the system. Each radiator connected to a standard $\frac{3}{4}$-in. main circuit has about 20 per cent of the water flowing in the $\frac{3}{4}$-in. pipe diverted



Radiator location	Distance X, Ft
Same level as boiler with down-feed risers	2.5
First floor above boiler with upfeed risers	1.0
Second floor above boiler with upfeed risers	2.5

(*From "I = B = R Installation Guide No. 1." Used by permission.*)

FIG. 200. Piping connections between a radiator and a main circuit of a one-pipe forced-circulation hot-water heating system.



(*Trane Co.*)

FIG. 201. Induced-flow fitting for single-main forced-circulation hot-water heating system.

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[2] "One-Pipe Forced-Circulation Hot-Water Heating Systems for Buildings Having Heat Losses Not Exceeding 60,000 Btu per Hour," *I = B = R Installation Guide Number I*, Second edition, The Institute of Boiler and Radiator Manufacturers, 60 East 42nd St., New York 17, N. Y., 1947.

through it. The radiator connection is made, as in Fig. 200, by the use of two $\frac{3}{4}$ by $\frac{1}{2}$ by $\frac{1}{2}$ in. tees with a length X of $\frac{1}{2}$-in. pipe located in the main between them. The table of values of X in Fig. 200 indicates the lengths of $\frac{1}{2}$-in. pipe required with various locations of the radiator with respect to the main circuit. Special fittings, as shown by Figs. 201 and 202, may provide for proper water circulation through a radiator without employment of the piping details of Fig. 200. The flow fitting shown in Fig. 201 is designed for placement in the water main at the point where the water passing through a radiator is returned to it. When this type of flow fitting is used at the point of return to the main, a standard tee is used at the point where a portion of the water flowing is to be diverted through a radiator or convector. Fig. 202 shows another type of flow fitting designed for placement in the main at the entering end of the radiator branch circuit. When a flow fitting of this nature is placed at the entering end of the radiator circuit, a standard tee is used at the point where the water flowing through the radiator is returned to the main. For a radiator that is below the level of the main, it may be

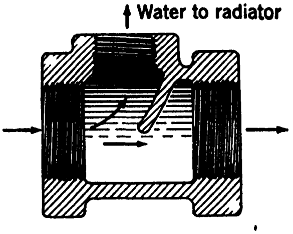


Fig. 202. Water distributor, one-pipe forced-circulation hot-water heating system.

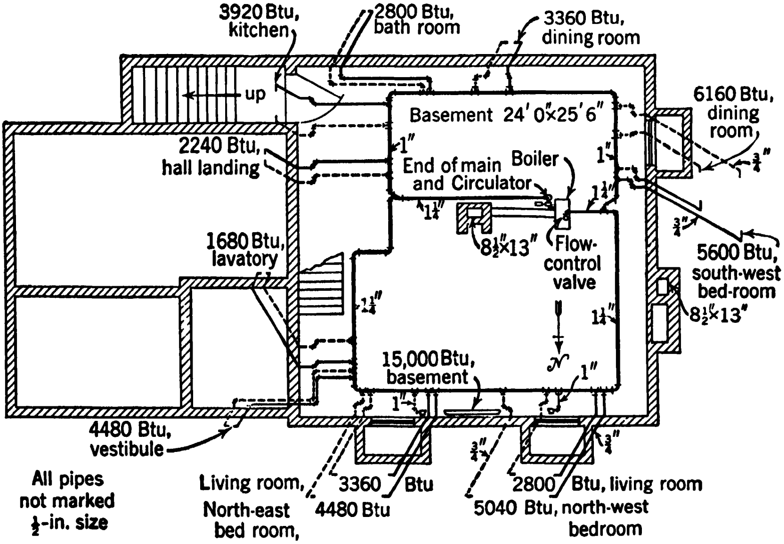


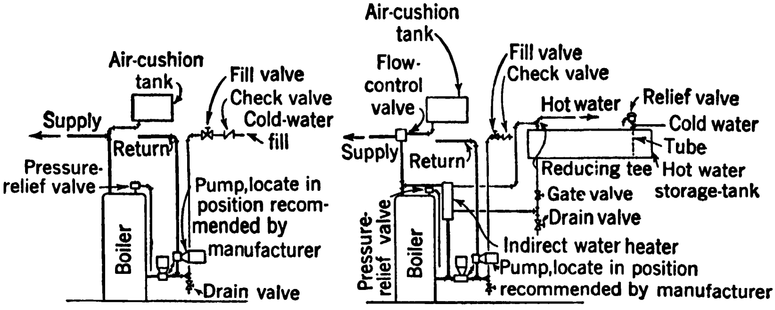
Fig. 203. Layout of basement piping for a one-pipe forced-circulation hot-water heating system in the I=B=R Research Home, Urbana, Illinois.

advisable to replace both tees with the proper types of flow fittings because in this case an adverse gravity head is developed in the risers. The system of Fig. 203 was designed for the use of flow fittings. Its design by use of the Guide would have resulted in radiator connections as shown by Fig. 200, 1-in. flow- and return-main connections to the boiler, and the two principal main divisions of $\frac{3}{4}$-in. pipe. A single-main circuit of $\frac{3}{4}$-in. pipe can be used in systems of the type under discussion when the number of radiators served does not exceed seven and the hourly heat losses are not more than 40,000 Btu.

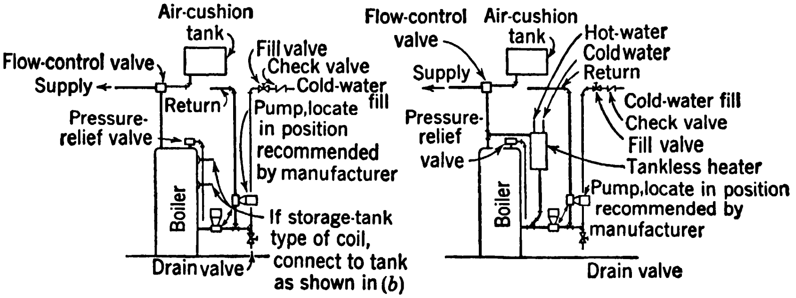
The guide design procedure, for applicable installations, consists of five simple parts which are:

1. An estimation by a reliable method, as given in Chap. 4, of the structure heat losses in Btu per hour. As a matter of convenience the guide includes tables of heat-loss factors for use in these calculations.
2. A determination of the square feet of edr required in each room for the given operating conditions. For simplicity the average temperature of the water in the radiators is assumed to be 10 F less than that of the water leaving the boiler, Heat-emission factors, applicable to various hot-water radiator operating conditions, are obtainable by the methods given in Art. 157. The required edr is equal to the heat losses of a room divided by a correct radiation factor.
3. The assumption is made that the total hourly heat losses of the structure are equal to the sum of the heat losses estimated for the individual rooms. The required number of water circuits is based on the total heat losses.
4, The boiler selection is made on the basis of a net I=B=R rating (see Art. 177), which indicates a capacity in Btu per hour either equal to or greater than the heat requirements of the building. Recommendations are that no extra allowance be made for the heating of a domestic water supply unless a residence has more than two bathrooms or the demand exceeds 75 gal per 24 hr.
5. The pump size is estimated by allowing one gallon per minute of water for each 10,000 Btu per hr of total heat losses. The operating head of the pump is arbitrarily taken as 5 ft of water, as practically all installations to which this method is applicable have been found to have a total resistance to flow less than the assumed amount. The guide includes a method of calculating the total head to be overcome in a system of this sort, and such a procedure should be used when more than 7 radiators are attached to a main circuit. The guide also includes a more exact method of radiator selection where consideration is given to the decrease of the water temperature as it flows in the circuits and the water-temperature variations in the radiators.

Typical boiler-piping connections are shown for 4 different domestic water-heating arrangements by Fig. 204. A flow-control valve is always required except where the boiler is not to be used for domestic water heating, and even then its use might be desirable from the standpoint of temperature regulation in the buildings. The function of this special valve (Fig. 205) is to prevent flow of water through the system due to gravity action. The small gravity head is insufficient to cause the valve to open, but it does not offer a serious resistance to flow when the circulating pump is in operation.



(a) Without domestic hot-water supply (b) With indirect heater and storage-tank installation



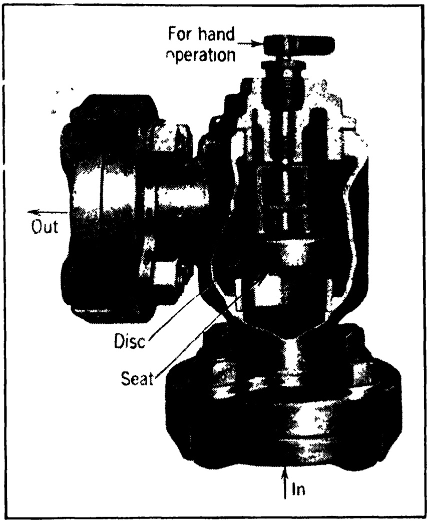
(c) With submerged heater installation (d) With indirect tankless installation

(*From "I = B = R Installation Guide No. 1." Used by permission.*)

FIG. 204. Typical boiler-piping connections in single-main forced circulation hot-water heating systems.

258. Hot-Water Heating by Use of Electrical Energy. Space heating by drawing on a supply of electrical energy at any time is likely to prove very expensive unless light loads are involved and the cost of the power supply is relatively low. Therefore such use of electrical energy may be justified only for special services during short periods of time. Present-day electrical heating equipment includes electrically operated portable steam radiators, unit heaters, and direct radiant heaters using either luminous or non-luminous resistance coils. These units produce heat when it is required and do not involve the storage of heat energy within any medium.

The load on the average electrical central-station power plant fluctuates during 24-hr periods, depending upon the class of service rendered, and usually involves peak demands at various intervals during the period. Attempts at space heating have been made by using power from a central station during hours of off-peak load. The most common scheme is to heat water by means of some form of immersion electrical heater units inserted into large storage tanks located within the building to be served. Heated water drawn from these tanks may be mechanically circulated through the piping of a hot-water heating system. The objective of the water-filled tanks, in which the heat is stored, is to make use of electrical energy which is available only during certain periods of each day and which may be had at a lower cost than power which may be drawn from a central station during the hours of peak demand upon the plant. The problems of control in such a system involve those of temperature regulation and safety precautions.



(*Bell and Gossett Co.*)

FIG. 205. Flow-control valve.

PROBLEMS

1. An open-tank gravity-flow hot-water heating system has 400 sections of small-tube direct cast-iron radiation, which are four tubes wide and 25 in. high. Compute the dimensions, in inches, of a required cylindrical expansion tank.

2. Calculate the dimensions, in inches, of a cylindrical expansion tank for an open-type hot-water heating system which has 5000 lin ft of $1\frac{1}{4}$-in. steel pipe employed as direct radiation.

3. A closed-tank hot-water heating system has 120 sections of 5*A* wall radiation. The tank is subjected to a head of 20 ft of water which is maintained at an average temperature of 180 F in the various pipes. The system was originally filled with water at 55 F, and the barometric pressure is 28.8 in. of mercury. Find the required capacity of the tank in gallons.

4. What effect would the maintenance of 5 psig water pressure at the top of the highest radiator have on the size of the expansion tank of problem 3 if all other conditions are unchanged?

5. A hot-water heating system has 7000 sq ft of small-tube equivalent direct cast-iron radiation operating under the following average temperatures: water inlet, 200 F; water outlet, 170 F; and room air, 65 F. A U. S. gallon contains 8 pints and 231 cu in. How many times is the water circulated per hour?

6. A finned-tube radiator is of 2-in. pipe size and has 100 ft of linear length. The unit is operated to emit 10,000 Btu per hr when the water enters it at 180 F and leaves at 160 F. Compute the average velocity of water flow, in feet per second, if the pipe is continuously joined together by return bends without the use of manifolds.

7. A gravity-flow up-feed hot-water heating system has first-floor radiators located 8 ft above the mean height of the water heater. The second-floor radiators are 10 ft above those of the first story. The supply is delivered to the radiators at an average temperature of 190 F and leaves them with an average temperature of 170 F. Calculate the motive head in milinches for both first- and second-floor radiators. Check by use of Fig. 190. What theoretical velocity of water flow exists in each case?

8. A gravity-flow hot-water radiator receives water at 200 F and discharges it at 180 F. An actual velocity of flow of 0.5 fps is desired in the supply riser. The actual velocity of flow may be taken as 0.4 of the theoretical. How many feet above the mean height of the water heater must the radiator be placed?

9. A 2½-in. steel pipe rises three feet above a boiler outlet, has 62 ft of additional length, and six 90-deg short-radius elbows before it again enters the return tapping of the heater. Calculate the total friction losses in the circuit when 24,000 lb of water are handled each hour.

10. A radiator is listed as having 120 sq ft of edr and gives off 160 Btu per hr per sq ft when water enters the unit at 185 F and leaves it at 165 F. The friction losses in the radiator circuit beginning at the take-off from the supply main and ending at the point of connection to the return main are not to exceed 250 milinches. The supply connection has 5 ft of pipe, one angle radiator valve, 3 short-radius 90-deg elbows, and the radiator. The return is comprised of 5 ft of pipe, four 90-deg elbows, and the connection to the return main. Find the required sizes of and lengths of pipe in the supply and return connections so that the friction losses in the circuit will not exceed the specifications, when resistances in tees of the supply and return mains are not included.

11. What horsepower would be required for the pump in a heating system delivering 400,000 Btu per hr with a temperature drop of 20 F if the total friction loss in the system is 9 ft of water? The pump is to be placed in the return main where the temperature of the water under design conditions is 180 F. Assume an efficiency of 65 per cent. From Fig. 199 select the pump that is best adapted to this installation.

12. Find the pipe sizes for the gravity-flow system of Fig. 197 if the allowable drop of water temperature in the radiator is from 180 to 160 F when the radiator emits 10,000 Btu per hr. The lengths of the pipes together with the pipe fittings are as in the example of Art. 254.

CHAPTER 12

PANEL HEATING

259. Definition of Panel Heating. The heating of rooms intended for human occupancy by means of large warmed panels is referred to as panel heating or radiant heating. The term panel heating is the one preferred by the authors because the other one implies that all the heat output of such a system is in the form of radiant heat which is not the case. A heating system may be classified as a panel type when the heating medium, which may be either electricity, water, or air, is circulated through either wires, pipes, or ducts incorporated in large panels forming sections of the floor, ceiling, or walls. Heat is transferred by radiation from each warmed panel to all objects within the room and to all room surfaces except the one in which it is located. The air in the room is warmed by convection currents passing over the surface of the panel or over the surfaces of objects receiving radiant heat from the panel.

260. Fundamentals of Panel Heating. It has been pointed out in Chap. 3 that the human body loses heat to its environment by convection, radiation, and evaporation. In the effective temperature range which produces bodily comfort the loss of heat by evaporation is comparatively small and nearly constant over a considerable range of dry-bulb temperatures and relative humidities. Therefore under comfort conditions the heat loss from the body is chiefly related to the combined effect of convection and radiation. The rate of heat loss by convection depends upon the average temperature difference between the surfaces of the exposed skin and outer clothing and the surrounding air, the area of such surfaces, and the velocity of the air movement over them. The rate of heat loss by radiation depends upon the area of the exposed skin or clothing surfaces and upon the difference between the average temperature of the surfaces and the mean radiant temperature of the surrounding walls, floors, ceilings, or objects. The mean radiant temperature may be defined as a uniform temperature of all the surrounding surfaces which would result in the same loss (or gain) of heat by radiation as that to (or from) the same objects and surfaces at their actual surface temperatures. The same feeling of comfort can therefore be produced with a comparatively high mean radiant temperature and a comparatively low air temperature, or vice versa. However, in the operation of a panel heating system, it is not possible to control the mean radiant temperature and the air temperature independently.

Under comfortable still-air conditions during the heating season the mean of the skin and outer clothing temperatures of persons normally clothed is approximately 81 F. Since there are considerable portions of the body such as the insides of the arms and legs which radiate their heat to other body surfaces, the effective surface exposed to loss by radiation is less than the effective surface exposed to loss by convection. The average adult person has an effective surface exposed to radiation amounting to 15.5 sq ft and a total surface exposed to the loss of heat by convection amounting to 19.5 sq ft. As has been pointed out in Chap. 3, the normal rate of heat production in an average-sized sedentary individual is about 400 Btu per hr of which approximately 100 Btu per hr are rejected in the form of vaporization of insensible perspiration on external skin areas and evaporation into the air that is taken into the body through the process of respiration. The remainder, or approximately 300 Btu per hr, must be absorbed by the combination of loss by radiation and loss by convection. In an environment at 70 F the loss by convection from the clothed body amounts to approximately 140 Btu per hr, and the remaining 160 Btu per hr must be dissipated by radiation to all surfaces to which the body is exposed. If the mean radiant temperature of the surrounding envelope were raised to 81 F, which is the average surface temperature of a fully clothed man, the loss of heat by radiation would be reduced to zero, and the air temperature would have to be reduced to the point where loss of heat by convection alone would amount to 300 Btu per hr. The still-air temperature that would be required to produce comfort for the individual under these conditions is approximately 60 F.

In the actual application of panel heating, a certain portion of the envelope surrounding a room, such as either the floor or the ceiling, is heated to whatever temperature is required to maintain the desired effect. The temperature of the remainder of the envelope and the temperature of the air in the room will vary with changing weather conditions. Early designers thought it possible to raise the mean radiant temperature of the envelope of a room, by means of heated panels, to a value considerably above that which would prevail when other means of heating are used, thereby producing comfort conditions for people with a considerably decreased room-air temperature. However, experience has shown that convection currents over the surfaces of the heated panels result in the heating of the air in the room to the point where it approaches the temperature maintained by other systems of heating. Consequently the relationships of convection and radiation in a room heated by large warmed panels are not greatly different from those found in rooms heated by either warm air or warm water in conventional heating plants.

Studies of the conditions affecting comfort have indicated that the average lightly clothed adult at rest is comfortable in an air temperature of 70 F when the mean radiant temperature of the surrounding envelope is at the same temperature. It has also been ascertained that for every degree Fahrenheit that the mean radiant temperature of the envelope is increased, the air temperature may be decreased one degree Fahrenheit. A simple expression called the comfort equation is

$$t_a = 140 - MRT \tag{115}$$

where t_a = air temperature required for comfort, deg F.
MRT = mean radiant temperature of the surrounding envelope, deg F.

Design of radiant panels on a rational basis involves the simultaneous solution of several heat-balance equations and requires a considerable amount of laborious computation. The problem is greatly simplified by regarding the heated panel as the source of heat input to the room, in which case it is usually assumed that the inside-air temperature will be maintained at 70 F. When panels are designed from this simplified approach, the procedure is quite similar to that followed in designing systems using conventional radiators. Assuming an inside air temperature of 70 F results in the assumption of a temperature of the heating medium which is somewhat greater than necessary, because the slight increase in the mean radiant temperature of the room due to the heated panel will permit comfort conditions to be maintained with an air temperature that is slightly lower than 70 F. However, this slight error in the design procedure causes no difficulty from an operating standpoint, provided that the temperature of the heating medium that is circulated through the panel can be properly controlled.

261. Advantages of Panel Heating. Heating panels which are incorporated in the floor, ceiling, or walls of a heated room appear to have the following advantages over the ordinary steam, hot-water, and warm-air heating systems.

1. The heat disseminating elements are completely out of sight and do not interfere with any decorative plan.

2. The heating panels do not interfere in any way with any desired placement of rugs and furniture.

3. Since the panels are incorporated in the structural frame of a building, the heating plant may be started as soon as this portion of the construction has been completed, thus facilitating the completion of interior work when construction is carried on during cold weather.

4. When the heated panels are either in the floor or in the ceiling this type of heating system does not require consideration in the placing of interior partitions and does not interfere with later moving of such partitions. This advantage is of greatest value in office buildings where it may be desirable to rearrange interior partitions from time to time.

5. Heating panels in any of the three possible locations produce warmer floors than any of the commonly used types of heating systems.

6. Heating panels are so arranged that it is impossible for ignorant or mischievous persons to throw the system out of adjustment by tampering with valves or dampers.

7. Air currents within a room heated by panels are of lower velocity than in rooms heated by conventional means with the result that dust particles to which disease-producing organisms may be attached can settle more readily.

8. Heating panels may also be used for radiant cooling where warm-weather relative humidities are consistently low.

9. Large heating panels operated at comparatively low temperatures do not include any dust-collecting surfaces and do not cause streaking of interior decorations, thus reducing both the cost of cleaning and the required frequency of redecoration.

10. It is believed by some, who have studied the physiological and psychological effects of panel heating, that the air in a room heated by this system possesses an intangible superior quality when compared with air in a room heated by other means, because it has not been "baked" by passing over hot metal surfaces.

262. Disadvantages of Panel Heating. Before deciding to install a panel-heating system, instead of one of the available older types, a prospective customer should weight its disadvantages which follow against the advantages which have been set forth in the preceding article.

1. Because of the heat storage in the large panels, which are used as heat disseminators, the problem of maintaining uniform room comfort conditions through rapid changes in the outdoor air temperature is usually more difficult than with other heating systems. Considerable trouble in this respect has been experienced in buildings where heavy floor panels have been combined with building construction

which incorporates a large portion of glass area in the walls. The problem of controlling heating plants using large panels will be discussed more fully in Art. 268.

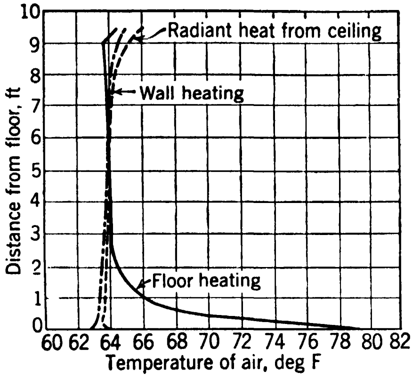
2. A temperature-control problem is inherent in panel-heating systems because the mean radiant temperature in a room heated by this means usually increases as the outdoor-air temperature decreases, thus requiring a reduction in the room-air temperature for the maintenance of comfort conditions. A similar control problem exists in buildings heated with conventional systems, particularly in buildings having large glass areas and in buildings in which the coefficient of heat transmission for the walls is high, as an increase in the room-air temperature is required to maintain comfort conditions in extreme cold weather. Although a control problem due to variation in the required air temperature exists with all types of systems, the problem of maintaining comfort conditions during a variation in outdoor temperature may be considerably more difficult to solve in some panel-heated buildings than in any building heated by conventional systems.

3. Heating panels are usually constructed on the job by the building or heating contractors and are not as readily adaptable to standardization as are the heat-dispensing units used in the conventional types of heating plants.

4. Panel-heating systems do not offer the possibility of supplying outside air for ventilation to the heated rooms through the system as do certain types of conventional systems. This is not likely to be a serious disadvantage when this type of system is being considered for residence heating except when groups of people are entertained in the home. However, in public buildings, it may be necessary to provide a separate system for ventilation when heating is to be provided by warmed panels.

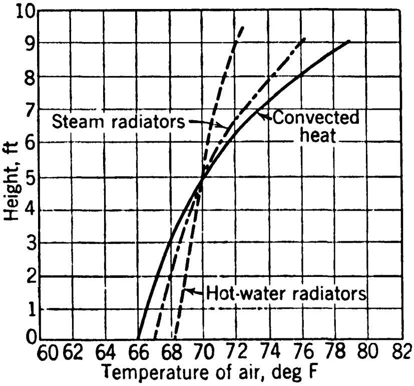
5. Heating panels are incorporated as an integral part of the building structure, and the repair of a leak which may develop after the system is placed in service can be very expensive.

263. Floor vs. Wall vs. Ceiling Panels. Heating panels which are incorporated in the building structure are limited to a low rate of energy dissipation, and it may be necessary to place them in more than one room surface because a floor, for instance, may not provide the required capacity. However, in general, sufficient surface may be installed in either the floor, the ceiling, or the walls. There is little advantage in one location over either of the other two from the standpoint of temperature gradient from the floor to the ceiling (see Fig. 206). From a comparison of Fig. 206 with Fig. 207 it may be noted that a radiant panel in any of the three possible room surfaces produces more uniform temperatures throughout the height of a room than the usual steam and hot-water radiators. As indicated by Table 86 the room surface chosen for the location of the panel has an important



(*A. M. Byers Co.*)

FIG. 206. Comparison of results with radiant heating from ceiling, wall, and floor.



(*A. M. Byers Co.*)

FIG. 207. Air temperatures at various heights above the floor for heating with steam and hot-water radiators and convection systems.

bearing on the operative heat-dissipating capacity of the unit in Btu per hour per square foot of exposed surface. Therefore the greater capacity of panels in ceiling and wall locations may cause one of these

TABLE 86

APPROXIMATE HEAT OUTPUT OF RADIANT PANELS UNDER DESIGN CONDITIONS

Room-air temperature assumed at 70 F; mean radiant temperature of other five room surfaces assumed at 65 F

Location of Panel	Maximum Allowable Surface Temperature, Deg F	Maximum Total Heat Output, Btu per Sq Ft per Hr	Maximum Heat Output by Radiation, Btu per Sq Ft per Hr	Maximum Heat Output by Convection, Btu per Sq Ft per Hr	Per Cent of Heat Output by Radiation	Per Cent of Heat Output by Convection
Floor	90	43	24*	19	55*	45*
Wall	110	65	42*	23	65*	35*
Ceiling	120	83	58*	25	70*	30*

* Data taken with permission from *Hot Water Heating and Radiant Heating and Radiant Cooling*, by F. E. Giesecke.

locations to be used in buildings having large heat losses under design conditions. If a floor location is strongly favored for other reasons, a panel in this location may be used by supplementing it with one or more smaller panels installed either in a wall or in the ceiling. However, from an economic standpoint, it is desirable to choose a location which will permit a panel of sufficient capacity to be installed in a single room surface.

Floor panels are more easily installed, particularly in basementless buildings where the network of pipes for circulating the heating medium is supported the proper distance above a fill of suitable material after which a concrete floor slab may be poured around the pipes. The cost of a heating panel in this location is less than those placed in either of the other two room surfaces. A majority of the heating panels which have been installed to date in the United States are incorporated in a concrete floor slab.

264. Steel Pipes vs. Copper Tubing. Heated water is the medium most generally used in panel systems now in use, and the conduits conducting it through the panels may be made from wrought-iron pipes, steel pipes, or copper tubing. Wrought iron is a special grade of steel so that the physical properties of the two first mentioned materials are almost identical.

Table 87 gives a comparison of the merits of either wrought iron or steel with those of copper, obtained from an impartial survey of claims made by each of the two interested industries.

TABLE 87

Comparison of the Advantages of Wrought-Iron or Steel Pipe and Copper Tubing When Used as Conductors of Hot Water in Panel Heating

Advantages of Wrought-Iron or Steel Pipe	Advantages of Copper Tubing
1. Lower cost for piping	1. Tubes are easier to bend on the job
2. Has practically the same coefficient of expansion as concrete	2. Copper is completely resistant to corrosion by any chemical compound that is likely to be found in the circulating water
3. Pipe walls are strong enough to resist damage in handling, in fabricating, and while waiting for slab to be poured	3. Tubing walls and joints are smoother, thus reducing friction losses from those existent in the same diameter of wrought-iron pipe handling the same volume of water
4. Welded and screwed joints in wrought-iron pipe are stronger than soldered joints used for connecting lengths of copper tubing	4. Soldered joints in copper tubing are easier to make than threaded or welded joints in wrought-iron pipe
5. Larger pipe diameters can be used in floor panels for the same expenditure, thus reducing pumping costs due to lower friction losses	5. Practicable to use smaller diameter tubing in ceiling panels, thus reducing the thickness of plaster required and consequently the thermal storage of the panel

Greater thermal conductivity is sometimes claimed as an advantage in favor of the use of copper, and greater emissivity of the bare pipe is likewise claimed as an advantage in favor of wrought-iron or steel, but, since the tubes or pipes are usually embedded in concrete or plaster, the emissivity of the bare pipe is not worthy of consideration, and the resistance of the pipe wall is a neglible factor in the transfer of heat from the water to the air of the room regardless of which of the two metals are used. Proponents of the use of copper point out that the material is perfectly capable of withstanding the compressive forces set up in a tube because its coefficient of expansion is greater than that of concrete. They also point out that the bond between the copper tubing is strong enough to withstand the strain on it and that the stress in the tube and the tendency of the force created to break the bond is not accumulative and therefore is not greater in long embedded tubes than in short ones. Likewise, the proponents of wrought-iron or steel pipe explain that the chances that conduits made of this material and used in a radiant panel will fail because of corrosion are very remote.

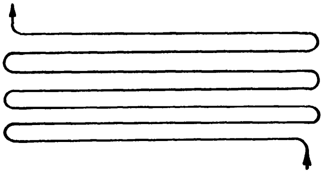


FIG. 208. Continuous coil.

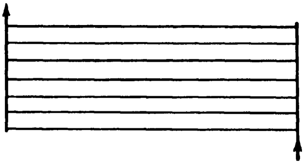


FIG. 209. Grid-type coil.

265. Coil Design. Pipe elements for heating panels are usually made in one of two basic patterns, namely, continuous coil as shown in Fig. 208 or grid as shown in Fig. 209. The continuous coil is usually less expensive to fabricate especially when copper tubing is used to form the conduit for conducting the water through the panel. However, the grid pattern has one important advantage in that the velocity of the water in any section of the unit is less, and consequently the friction losses are less when the water flows through adjacent pipes in parallel instead of in series as is the case when the pipe is laid in a continuous coil.

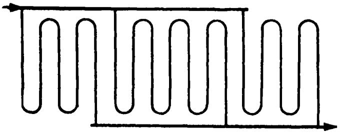
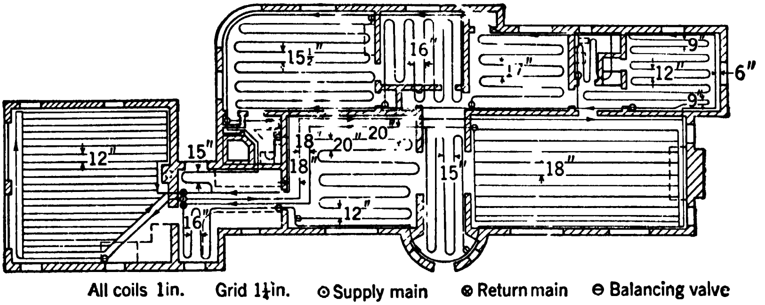


FIG. 210. Combination grid and continuous coil.

Figure 210 shows a pattern which may be used to combine several continuous coils of moderate length in a semi-grid pattern. This layout produces a network of pipes which is less expensive to fabricate than a grid pattern and one in which the frictional resistance to flow of the required amount of water is considerably less than that in a continuous coil providing the same total length of pipe.

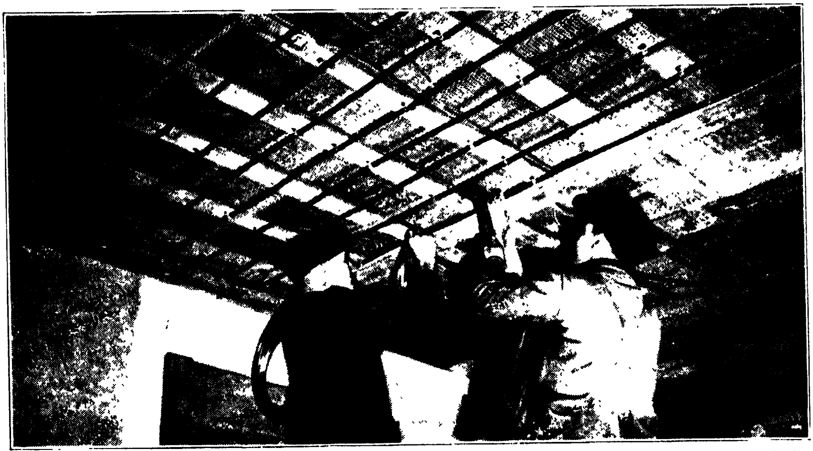
Figure 211 shows the complete layout of piping for a one-story basementless house which is to be heated by radiant floor panels. Note that both continuous coils and grids are used in this layout.



(*A. M. Byers Co.*)

Fig. 211. Typical radiant-heating piping.

Figure 212 illustrates the use of copper tubing in a ceiling panel. The coil shown is of the continuous type and is being attached to the under side of the metal lath. The cost of plastering is reduced when the tubing is placed on the other side of the lath. However, care must be taken to force the plaster through the lath to make a good contact with the tubing when in this position.



(*Chase Brass and Copper Co.*)

Fig. 212. Ceiling panel of copper tubing attached to expanded-metal plaster base.

266. Pipe Coils under Wood Floors. In the majority of heating panels which have been installed to date, the pipes or tubes have been embedded in either concrete or plaster, the heat being transferred by conduction from the pipe surface to the slab surface from which it is dissipated by a combination of radiation and convection. However, successful performances have been claimed for systems in which the pipes or coils were placed underneath wood floors, between joists in wood floor construction, or between the sleepers in wood floors placed on top of a concrete slab. When this type of panel construction is used, the wood floor becomes the radiant panel, heat being transferred from the pipes to it by radiation and convection through the air space between the joists or sleepers.

Experiences with several installations of this type indicate that wood flooring will not be damaged by the heat from the hot pipes, provided that the wood of which the floors are made has been thoroughly dried. It has been found that the heat transferred from the water to the air of the room is approximately 1.75 Btu per hr per sq ft of pipe surface per deg temperature difference and that the length of pipe required is approximately double that required when pipes are embedded in a concrete slab.

267. Air Vents and Expansion Tanks for Heating Panels. Air vents should be provided at all high points in a panel-heating system, if the circulation is to be by gravity flow. However, when a pump is used, the flowing water can be depended upon to carry the air to other points which may be more convenient for the locations of vents. Either automatic vents or manually operated compression cocks can be used, but all vents regardless of type should be installed in such a way that they are readily accessible. Coils, pipes, and mains should be arranged to reduce to a minimum the number of points requiring air vents.

The procedure to be used in sizing the expansion tank for a water-heated panel plant is the same as for any other hot-water system except that it is necessary to actually calculate the volume of the water involved. Lengths of different sizes of standard pipe containing one cubic foot of water are given in Table 68, Chap. 9. The volume of the boiler may be obtained from the manufacturer. Since the maximum water temperature occurring in panel systems is lower than that customarily used under design conditions in hot-water systems using radiators and convectors, some additional capacity will be provided in the expansion tank if the same allowance for expansion is made in its design. Expansion tanks in panel systems may be either the open or the closed type if properly placed as to elevation. The expansion tank should be connected to the return main on the suction side of the circulator and as close to it as practicable.

268. Control of Panel Heating. Probably the most difficult problem in connection with the design and installation of heating panels is the selection of the proper system for regulating their heat output. Improper selection of controls is the real reason for failure in the majority of cases where the systems have not been able to produce the degree of comfort which had been expected. Insufficient pipe surface in a panel can, to a considerable extent, be compensated by raising the temperature of the circulating water above that on which the design was based, but phenomena such as "flashing" and "hunting" due to an improper control are in some instances most difficult to handle. Heating panels in which pipes are embedded in concrete or plaster are certain to possess considerable thermal inertia and also considerable resistance to the flow of heat from the water to the surface of the panel. Building structures vary greatly in regard to thermal inertia because some have walls made of either one or more thin sheets of metal or wood products while others have walls including several inches of solid masonry. Building structures also vary greatly in regard to thermal resistance as some walls may contain several inches of insulation while others have none. Buildings having a large proportion of their wall areas taken up by windows or doors have very little thermal resistance.

Large thermal capacity and thermal resistance in heating panels cause slow responses to temperature controls, whereas large capacity and resistance in the structure produce delays in the reflection of external load changes within the building. Usually, but not always, such structural lag is advantageous from a control standpoint as it permits the use of a control element that is actuated by changes in the outdoor air temperatures for anticipation of the load changes.

The factor of greatest importance from the standpoint of control of a panel-heated building is the relationship between the thermal inertia of the panel and that of the structure. If the time required for an externally imposed load to be reproduced within the building is equal to the time required for a change in circulating-water temperature to produce a change in the surface temperature of the panel no serious control problem exists, and any properly designed control system using an outside temperature bulb will maintain comfort conditions within the structure even through a rapid change in the outdoor weather. However, when a building having very little thermal inertia, such as a house provided with an unusual amount of glass area, is heated with a panel having large thermal inertia, such as a concrete floor panel, maintenance of comfort conditions within the house through a sudden change in outdoor temperature is impossible with any control equipment that is available at the time of this writing. On the other hand, a concrete-floor panel in a house having concrete walls supplemented with some insulation to produce thermal resistance as well as thermal inertia could be controlled in a satisfactory manner. The best results are possible when the thermal inertia of the panel and that of the structure are equal. Satisfactory results can be obtained when the thermal inertia of the panel is less than the thermal inertia of the structure. However, the control of the panel is certain to be unsatisfactory when the thermal inertia of the panel is appreciably greater than that of the structure.

If the effect of the occupants and interior processes is neglected, the relationship between the air temperature and the mean radiant temperature of the six surfaces of a panel-heated room will depend entirely on the amount of outdoor air that is introduced by either infiltration or deliberate ventilation. With zero ventilation the air temperature would approximate the average temperature of the room surfaces, and an ordinary room thermostat set at 70 F could be used for the control of a panel system. However, when outdoor air is introduced at temperatures below 70 F such a room thermostat set at this temperature would demand more heat from the panel to warm this air resulting in an increase in the surface temperature of the panel and consequently an increase in the mean radiant temperature of the six room surfaces. It is therefore clearly evident that a panel-heated room in which the heating system is controlled by a room thermostat set at 70 F will maintain an environment that is too warm for perfect comfort at all times when outdoor air is introduced at a temperature below the thermostat setting. Introduction of outdoor air calls for a reduction in the thermostat setting, and the amount of the reduction is directly proportional to the amount of air introduced and inversely proportional to the temperature of this air.

In well-built homes requiring no mechanical ventilation, lightweight panels such as those made of small tubes with metal lath and plaster may be satisfactorily controlled with an ordinary room thermostat of good design. No noticeable discomfort is likely to result from the fact that the air temperature is slightly above that specified by the comfort equation during periods when the outdoor-air temperature is low. This slight overheating in cold weather can easily be eliminated if desired by reducing the thermostat setting 1 or 2 deg. However, in buildings the use of which requires the introduction of large amounts of outdoor air, a reduction in thermostat setting as much as 10 F may be required during extremely cold weather, and frequent adjustment of the thermostat setting may be required for the maintenance of comfort during periods when the outdoor temperature is changing. The obvious solution to the temperature-regulation problem in a building of this sort heated with a panel system is a control device that keeps the temperature of the inside air in the proper relationship to the outdoor-air temperature. In the knowledge of the authors, no such control system is available at this time.

TABLE 88

OPTIMUM INDOOR TEMPERATURE IN PANEL-HEATED ROOMS FOR DIFFERENT OUTDOOR TEMPERATURES AND DIFFERENT AMOUNTS OF VENTILATING AIR*

Ventilation Air, Cu Ft per Hr per Sq Ft of Room Surface	Outdoor Temperature, Deg F	Optimum Inside-Air Temperature with Ceiling Panels, Deg F	Optimum Inside-Air Temperature with Wall Panels, Deg F	Optimum Inside-Air Temperature with Floor Panels, Deg F
0	Any Temperature below 70	70	70	70
2	30	68.8	69.2	69.0
	20	68.6	68.9	68.8
	10	68.4	68.7	68.5
	0	68.0	68.5	68.2
	−10	67.6	68.3	68.0
	−20	67.2	68.0	67.7
4	30	67.7	68.4	68.0
	20	67.2	67.8	67.5
	10	66.7	67.5	67.0
	0	65.9	67.0	66.5
	−10	65.2	66.6	65.9
	−20	64.5	66.2	65.3
6	30	66.5	67.5	67.0
	20	65.9	66.7	66.3
	10	65.0	66.2	65.5
	0	63.9	65.6	64.6
	−10	62.8	65.0	63.7
	−20	61.6	64.3	63.0
8	30	65.3	66.7	66.0
	20	64.5	65.6	65.0
	10	63.5	64.9	64.0
	0	61.8	64.1	62.8
	−10	60.3	63.3	61.7
	−20		62.5	60.5

* Data compiled from graphs prepared by the engineering staff of the Minneapolis-Honeywell Regulator Company. Data are for outer walls having a U value of 0.12 and are valid for all walls of conventional good construction having U values from 0.05 to 0.20.

Table 88 compiled from data accumulated by the engineers of a prominent manufacturer of control equipment gives optimum indoor temperatures in panel-heated structures for different amounts of ventilating air, covering a considerable range of outdoor-air temperature and three different panel locations.

269. Design of Panel-Heating Systems Using Heated Water. Many different procedures have been proposed for the design of the panels in panel-heating systems, but there is at the present time no one method which has been generally accepted as standard. There are many factors which affect the performance of a heating panel, such as temperature of the panel and that of the unheated surfaces, emissivity of the various surfaces, air temperature, size of pipes, spacing of pipes, conductivity of covering material, and depth of cover. A completely accurate design procedure taking into consideration all of the factors is exceedingly involved and laborious. In all the methods which have been proposed for practical application, certain assumptions are made in regard to some of the less-important factors in order to simplify the solution. Some of the proposed methods include a mathematical analysis of each problem while others make use of a rather extensive series of charts. Either of the above-mentioned methods is beyond the scope of this book, and it is impossible in the space that can be allotted to this subject to cover all the possible arrangements which may be used. The two arrangements most commonly used are floor panels in which pipes are embedded in concrete as shown in section by Fig. 213 and ceiling panels in which either small pipes or tubes are attached to metal lath or expanded metal and are covered with plaster as shown in Fig. 214. Wall panels when used to supplement either floor or ceiling panels can be constructed in exactly the same manner as the ceiling panels.

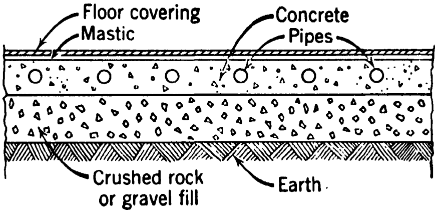


FIG. 213. Section of a concrete floor panel.

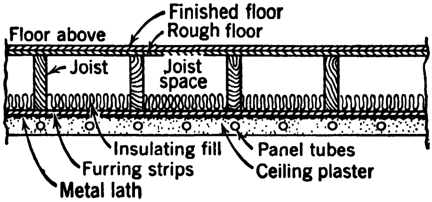


FIG. 214. Section of an insulated ceiling panel.

The general procedure to be followed in designing a panel-heating system is the same regardless of the details of the panel and is given in outline form. Detailed information for carrying out certain steps will be given only for the two types of panel construction which have been mentioned.

The design procedure which follows is an attempt to take into consideration all the factors which might seriously affect the performance of a panel in a method which is not too involved for use by the practical designer. Tables and charts have been drawn from the available literature of the day.

270. General Design Procedure for a Panel-Heating System.

Step 1. Compute by conventional methods (see Chap. 4) the hourly heat losses from each of the rooms to be heated, assuming for calculation purposes that the room air temperature will be maintained at 70 F (see explanation of step 1).

Step 2. Make a careful survey of the shapes and uses of the various rooms and decide upon the surface in each one which is to be used as the heating panel (see Art. 263).

Step 3. Divide the heat losses from each room by the area of the surface to be used to obtain the required output of each panel in Btu per hour per square foot of available area. A comparison of these required output rates will disclose the panel of maximum required capacity.

Step 4. For each room calculate the mean radiant temperature of the five unheated room surfaces, assuming for the moment that the heating panel will be confined to the surface selected (see explanation of step 4).

Step 5. Using the results from steps 3 and 4, obtain the required panel surface temperatures (see explanation of step 5).

Step 6. Estimate the heat losses from the reverse side of each panel in Btu per hour per square foot of panel surface, and add this quantity to the required heat emission rate from the surface facing the room.

Step 7. Select the allowable mean temperature of the circulating water (see explanation of step 7).

Step 8. Select a combination of pipe size, pipe spacing, and depth of cover which will provide the required total unit output, first for the panel of greatest demand then for the other panels, using the same mean temperature of the circulating water that was calculated for the panel of the greatest unit heat output (see explanation of step 8).

Step 9. Sketch the proposed coils on the plan of the building to be heated (see Fig. 211).

Step 10. Lay out a system of supply and return mains servicing all the individual panels.

Step 11. Calculate the weight of water which must be circulated through each panel, and estimate the friction loss using the method that is outlined in Art. 250.

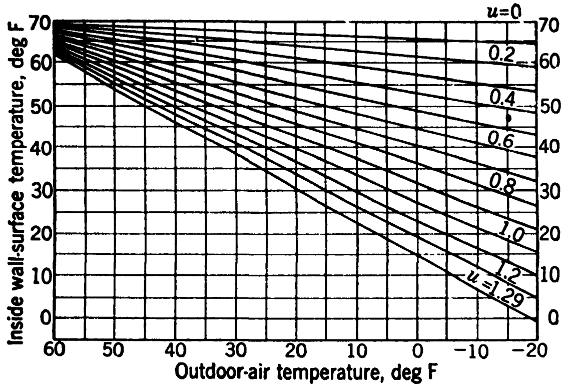
Step 12. Estimate the friction losses in the entire system.

Step 13. Select the boiler and the circulating pump.

Step 14. Calculate the required size of the expansion tank, and indicate its proper location.

Step 15. Indicate the location of balancing valves for properly distributing the heated water to the various panels in the system.

Explanation of step 1. If the panel in any room is to be located in an exposed surface, such as the floor in a basementless house, the room heat loss through this panel surface is assumed to be zero. Assumption of an air temperature of 70 F under design conditions will provide a factor of safety as the actual operating air temperature under design conditions will be somewhat less than 70.



(*From "Hot-Water Heating, Radiant Heating, and Radiant Cooling," by F.E. Giesecke. Used by permission.*)

FIG. 215. Inside-surface temperatures of outside walls.

Explanation of step 4. The mean radiant temperature of the five unheated surfaces in a room in which a heating panel occupies the sixth surface may be determined by the following procedure.

a. Calculate the area of each of the five unheated surfaces.

b. Determine the inside-surface temperature of all exposed walls, floors, and ceilings under design conditions. Use Fig. 215, and assume that the surface temperature of all interior partitions is 70 F. The U values for the different exposed walls have previously been determined in the calculation of step 1.

c. Multiply each of the several surface areas by the proper heat emission rate, in Btu per hour per square foot, from Table 89, and add the products to find the total heat emitted by all these surfaces. Divide this total by the sum of the five room-surface areas to find the *mean unit heat radiation* of the room surfaces exclusive of the one to be used as a panel. The mean radiant temperature of the five surfaces may then be found from Table 89 as that corresponding to the mean heat emission rate. Use of Table 89 requires reference to Table 90 in the determination of emissivity values for different types of surface finishes.

TABLE 89

HEAT EMISSION BY SURFACES HAVING VARIOUS TEMPERATURES AND EMISSIVITY FACTORS, BTU PER HOUR PER SQUARE FOOT*

Surface Temperature, Deg F	Emissivity Factor e					Surface Temperature, Deg F	Emissivity Factor e				
	1.00	0.95	0.90	0.85	0.80		1.00	0.95	0.90	0.85	0.80
20	91.8	87.2	82.6	78.0	73.5	70	136.0	129.3	122.3	115.6	108.8
22	93.4	88.7	84.0	79.4	74.7	72	137.9	131.0	124.0	117.2	110.3
24	94.9	90.2	85.4	80.7	75.9	74	140.2	133.1	126.1	119.2	112.1
26	96.5	91.7	86.9	82.0	77.2	76	142.8	135.7	128.5	121.4	114.2
28	98.1	93.2	88.3	83.4	78.5	78	144.9	137.7	130.4	123.2	115.9
30	99.7	94.7	89.8	84.7	79.8	80	147.2	139.9	132.5	125.1	117.9
32	101.4	96.3	91.2	86.2	81.1	82	149.3	141.8	134.4	126.9	119.4
34	103.0	97.9	92.7	87.6	82.4	84	151.5	143.9	136.4	128.8	121.2
36	104.7	99.5	94.2	89.0	83.8	86	153.8	146.1	138.4	130.7	123.0
38	106.4	101.1	95.8	90.4	85.1	88	156.0	148.2	140.4	132.6	124.8
40	108.0	102.8	97.2	91.8	86.4	90	158.5	150.5	142.7	134.7	126.9
42	109.9	104.4	98.9	93.4	87.9	92	160.6	152.6	144.6	136.5	128.5
44	111.6	106.0	100.5	94.9	89.3	94	163.0	154.8	146.7	138.6	130.4
46	113.3	107.7	102.0	96.3	90.8	96	165.3	157.1	148.8	140.5	132.3
48	115.2	109.5	103.8	97.9	92.3	98	167.7	159.3	150.9	142.5	134.2
50	116.9	111.0	105.3	99.4	93.6	100	170.3	161.7	153.2	144.8	136.2
52	118.8	112.9	106.9	101.0	95.1	105	176.3	167.5	158.7	149.9	141.0
54	120.6	114.6	108.6	102.5	96.6	110	182.3	173.2	164.2	155.0	146.0
56	122.6	116.4	110.3	104.2	98.1	115	189.1	179.7	170.2	160.7	151.3
58	124.4	118.2	112.0	105.7	99.6	120	195.6	185.7	176.1	166.3	156.5
60	126.3	119.9	113.8	107.4	101.1	125	202.6	192.5	182.4	172.2	162.1
62	128.2	121.8	115.3	109.0	102.6	130	210.9	200.4	189.8	179.3	168.8
64	130.1	123.5	117.1	110.6	104.1	135	216.8	206.0	195.1	184.3	173.5
66	132.1	125.5	118.8	112.3	105.8	140	224.1	212.9	201.8	190.5	179.2
68	134.0	127.3	120.5	113.9	107.2	150	238.0	226.1	214.4	202.3	190.5

* From *Hot Water Heating, Radiant Heating, and Radiant Cooling,* by F. E. Giesecke. Used by permission.

It may be noted from Table 91 that the emissivity *e* for all materials commonly used in room surfaces is close to 0.90 so that in most cases this value may be assumed for all the surfaces without the introduction of serious error.

Explanation of step 5. Authorities agree that approximately 70 per cent of the heat is delivered by radiation from a ceiling panel, 65 per cent from a wall panel, and 55 per cent from a floor panel. Therefore the heat to be delivered by radiation from each panel is the proper percentage, depending upon the orientation of the panel, times the total heat loss from the room to be heated, as calculated in step 1.

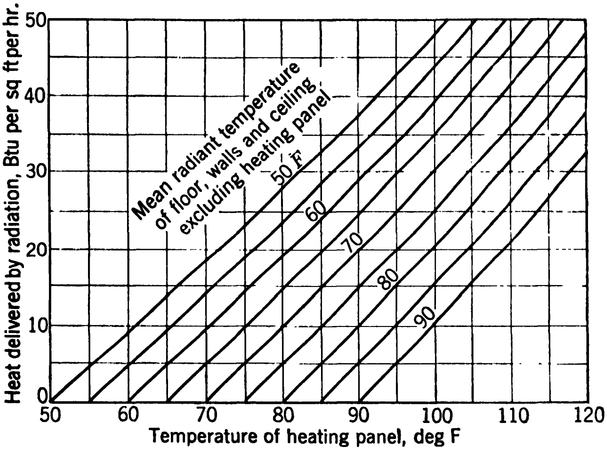


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FIG. 216. Heat delivered to rooms by radiation from panels.

Dividing the heat in Btu per hour that is to be delivered by radiation from the panel by its area will give the required output by radiation in Btu per hour per square foot of surface. Figure 216 can then be used to determine the required panel surface temperature.

TABLE 90

APPROXIMATE EMISSIVITY FACTORS FOR VARIOUS TYPES OF SURFACES AT A TEMPERATURE OF 70 F

Black body	1.00	Marble, polished	0.93
Aluminum, dull	0.22	Oak	0.90
Aluminum, polished	0.04	Oil paint	0.94
Asbestos, board	0.94	Paper	0.93
Brass, dull	0.22	Plaster	0.91
Brass, polished	0.03	Roofing paper	0.91
Concrete	0.90	Rubber, hard	0.91
Glass	0.92	Rubber, soft	0.86
Iron and steel, dull	0.82	Tile	0.90
Lead, dull	0.28	Water	0.95

Explanation of step 6. For a concrete panel laid on the ground the heat loss through the reverse side of the panel has been found to be small, provided that a suitable fill consisting of at least 6 in. of crushed rock or gravel is placed between the lower side of the panel and the ground and, further, provided that a strip of insulating material at least 1 in. in thickness is placed between the edges of the panel and the foundation walls of the building. If these provisions are made against excessive conduction of heat to the ground, an allowance of 10 per cent of the heat losses of the room has proven to be adequate.

For ceiling and wall panels, the loss of heat from the reverse side of the panels can be reduced to approximately 10 per cent of the heat delivered to the rooms, provided that at least 3 in. of rock wool or equally effective insulation is placed in the stud or joist spaces on the rear sides. If for any reason, such as the desire to deliver some heat to a room on the opposite side of a panel, insulation is not provided, the total resistance to heat flow in each direction from the plane of the pipes or tubes should be calculated and the reverse heat flow estimated by multiplying the amount of heat to be delivered to the room by the inverse ratio of the resistances. For example, if the resistance between the plane of the pipes or tubes and the reverse side is 4 times the resistance between this plane and the panel surface, the losses from the reverse side would be estimated as one-fourth of the heat required to heat the room.

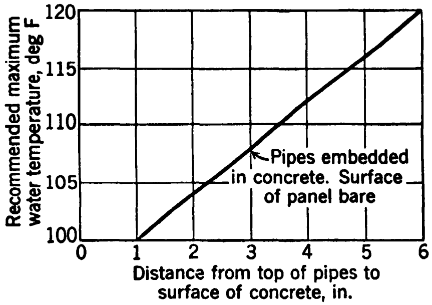


FIG. 217. Design water temperatures for concrete-floor panels.

Explanation of step 7. The higher the mean temperature of the circulating water the lower is the number of square feet of pipe required, but excessively high temperatures may cause cracking of the concrete or plaster and uneven heating at the surface. The recommended mean water temperature for design purposes is 130 F for pipes embedded in plaster in a ceiling panel and from 100 F to 120 F for pipes embedded in concrete in floor panels (see Fig. 217).

Explanation of step 8. In making a selection of the pipe size, pipe spacing, and depth of cover for the panel of greatest demand, it is necessary to assume a combination and then determine the mean temperature of circulating water that would be necessary with that combination to give the required rate of heat transfer from the water to the panel surface. If the required water temperature is too high, a combination must be selected which will provide more pipe surface per square foot of panel, unless it is practicable to decrease the depth of cover.

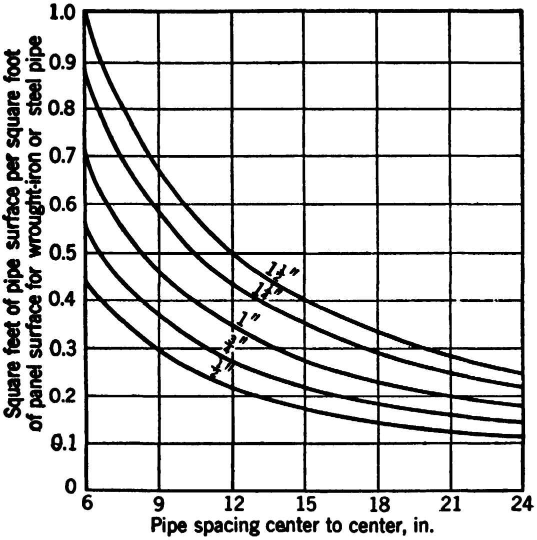


FIG. 218. Wrought-iron pipe surface per square foot of panel surface.

The heat conducted from the pipes to the panel surface per square foot of this surface may be represented by the following equation.

$$H = SR(t_w - t_p) \qquad (116)$$

where H = heat to be conducted or the total heat loss from the room divided by the panel surface area, Btu per hr per sq ft of panel surface.

S = pipe surface per square foot of panel, sq ft (see Figs. 218 and 219)

R = rate of heat conduction from pipe surface to panel surface per degree Fahrenheit of water to panel-surface temperature difference, Btu per hr per sq ft (see Figs. 220 and 221).

t_w = mean temperature of water, deg F.

t_p = temperature of panel surface, deg F.

Example. Specify the pipe size, pipe spacing, depth of cover, and mean temperature of the circulating water for a bare concrete floor panel which is to deliver a total of 30 Btu per hr per sq ft of panel surface. The mean radiant temperature of the other five room surfaces has been estimated as 68 F under design conditions.

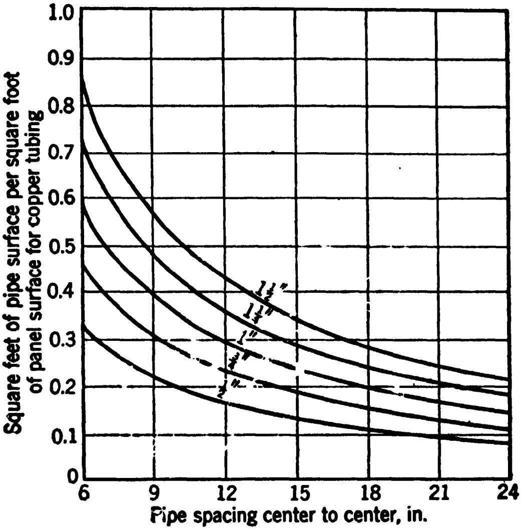
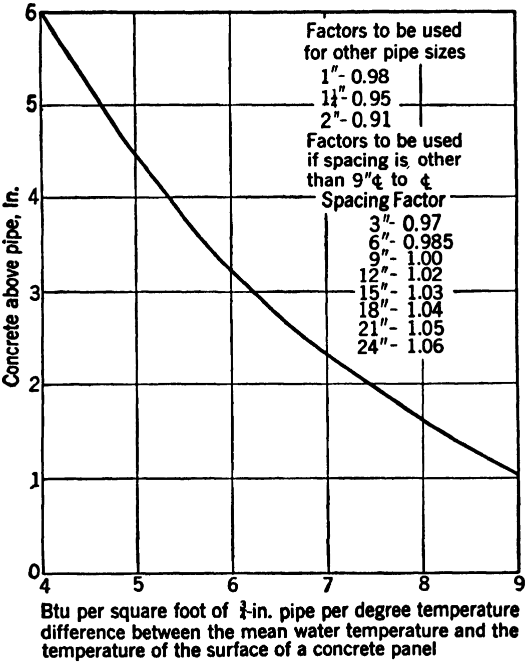


FIG. 219. Copper-tube surface per square foot of panel surface.

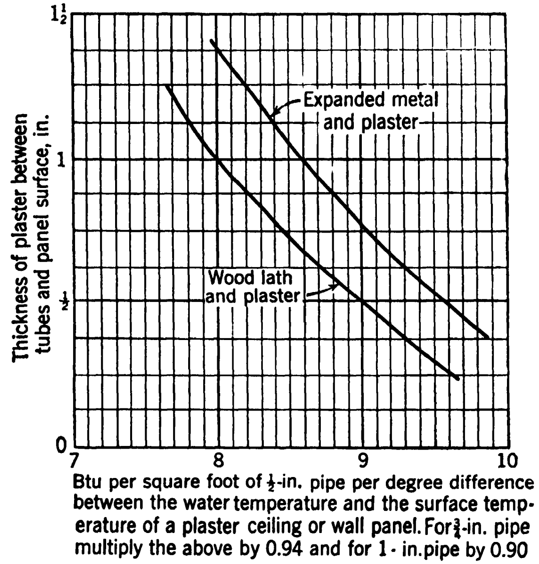


(*From* "*Radiant Heating,*" *by T. Napier Adlam. Used by permission of The Industrial Press, New York.*)

FIG. 220. Heat conduction per unit temperature difference for ¾-in. pipes spaced 9 in. apart and buried in concrete at various depths.

Assume that this panel is the one from which the demand is maximum for the system.

Solution. Since 55 per cent of the heat from a floor panel is by radiation the heat required by this means is 0.55 × 30 = 16.5 Btu per hr per sq ft. From Fig. 216 it may be determined that a panel surface temperature of 85 F will be required. The total required heat output of 30 Btu per hr per sq ft must be conducted from the pipes to the panel surface because of the difference between the mean water temperature and 85 F. Assuming a combination of $\frac{3}{4}$-in. pipe, 9-in. spacing, and 3-in. depth of cover, equation 116 may be rearranged to find the required mean temperature of the circulating water, as $t_w - 85 = 30/SR$. From Fig. 218, $S = 0.37$, and, from Fig. 220, $R = 6.3$. Therefore $t_w = 30/(0.37 \times 6.3) + 85 = 97.85$ F.



(*From "Radiant Heating," by T. Napier Adlam. Used by permission of the Industrial Press, New York.*)

FIG. 221. Heat conduction per unit temperature difference for $\frac{1}{2}$-in. pipes in a plaster panel.

According to the data of Fig. 217 the maximum allowable water temperature for this panel construction is 107 F. Although some economy in first cost could be effected by using less pipe and a higher temperature, the above combination would allow for the possibility that a floor covering may be placed over the bare concrete, thereby requiring an increase in the water temperature. Having determined the mean temperature of the circulating water for the panel of maximum demand, this temperature becomes fixed for all of the other panels, and it remains only to select a combination of pipe size, pipe spacing, and depth of cover which will deliver the required output.

Example. Assume that another panel in the system of the preceding example is required to deliver 28 Btu per hr per sq ft and that it has been determined through the same procedure that a surface temperature of 80 F is required. Assuming the same depth of cover that was used in the panel of maximum demand, select a suitable combination of pipe size and spacing.

Solution. The mean water temperature has been fixed by the calculation made for the panel of maximum demand at 97.85 F. R will also be the same, 6.3 Btu per sq ft per deg temperature difference, if the spacing factor is temporarily neglected. Substitution in equation 116, with the terms rearranged, gives $S = 28 \div 6.3(97.85 - 80) = 0.245$ sq ft. of pipe surface per sq ft of panel surface. From Fig. 218 it is apparent that $\frac{3}{4}$-in. pipes spaced on 13-in. centers will provide the necessary pipe surface for the panel. Referring again to Fig. 220 it may be noted that the spacing factor for pipes placed 13 in. apart is 1.023 instead of 1.0 as was assumed. Consequently 13-in. spacing provides about 2 per cent more pipe surface than is required, but, if balancing valves are provided in the pipes supplying individual panels, no operational difficulties will result from this slight excess of pipe surface.

If it is desired to use other than $\frac{3}{4}$-in. pipe, the procedure is exactly the same except that a factor for pipe size is applied to the curve in Fig. 220. The same general procedure may be used in designing the coils for wall or ceiling panels or for a panel of any orientation using copper tubes instead of wrought-iron pipes. If copper tubes are employed, Fig. 219 is used instead of Fig. 218, and, for plaster-finished wall or ceiling panels, Fig. 221 is used in place of Fig. 220.

Explanation of steps 9 to 15. After the design of all individual coils has been completed the weight of water to be circulated through each one may be found by dividing the required total panel output from step 6, by 15, assuming a temperature drop of 15 deg. The weight of water to be handled by each section of the supply and return mains may be determined by adding the weights required by the individual coils serviced. The procedure to be followed in sizing the mains and in estimating the friction loss of the entire system is the same as outlined for hot-water systems in Chap. 11.

271. Use of Air as the Heating Medium in Panel Systems. Although panel-heating systems employing heated water as the circulating medium greatly outnumber those using warmed air, the possibilities of the latter type should not be overlooked. Panel systems in which heated air is circulated can in general be installed at lower initial cost and are free from the danger that a leak may develop and cause serious damage to any part of the building or furnishings. Inasmuch as it is not necessary to embed pipes or tubes in the substance used for constructing them, ceiling or wall panels which have to support only their own weight may be much thinner when air is used as the heating medium instead of water. A thinner panel results in less thermal inertia and less resistance to heat conduction from the circulating medium to the surface which is exposed to the room that is being heated. Because of reduced mass in the panels it is easier to achieve a satisfactory control of the room temperature when air is used in place of water. Another advantage in using heated air instead of heated water is that local hot spots in the panel are eliminated and there is less danger that the plaster will crack. The only serious disadvantages in using air instead of water are the lower density and the lower specific heat of this medium necessitating the circulation of a greater weight and a much greater volume for the same heat output. It is therefore likely that the cost of power for the operation of a system employing heated air will be greater than that for a water-heated panel system of equal capacity. However, if proper attention is given to duct and panel design, the power required to operate a warm air-panel system will not be prohibitive from an overall economy standpoint.

Heated air can be circulated through hollow tile under concrete floors or between the joists of wood floors, but the ceiling panel offers the possibility of achieving light weight with conventional lath and plaster or its equivalent. If wood joists are used in the ceiling construction it is necessary to provide a circulating space below the bottom of the joists. This may be accomplished as shown in Fig. 222. Plaster board attached to the lower edges of the ceiling joists forms the upper boundary of the air passage. Special hangers attached to the joists support steel rods which in turn are used to support the metal lath over which the plaster is spread. One method of supplying heated air, directing its flow through the panel and returning it to the furnace, is illustrated in Fig. 223. The baffles may be made of metal or other suitable material and must be attached to the plaster board before installing the lath. The furnace is the same type as is used in conventional forced-circulation warm-air heating systems. The fan used may also be the conventional type, but it must be capable of handling a greater volume of air. Ducts for conducting air from the furnace to the panel and from the panel to the furnace may be of the same construction as the supply and return ducts of convection systems. However, in the warm-air panel system, the supply ducts are usually located in outer walls with the return ducts in inner partitions, which is contrary to general practice in designing the other type of forced warm-air heating plant. If friction losses are to be comparable to

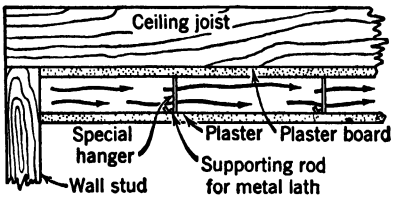


Fig. 222. Suspended warm-air panel.

those in a convection plant, both supply and return ducts must be made larger in proportion to the heat-carrying capacity in the panel type of installation. Figure 223 shows a furnace serving only one panel, but by means of suitable supply and return trunk ducts a single furnace can serve several panels in the same general manner in which a convection system supplies heated air to and returns recirculated air from several rooms in the same building.

The suspended ceiling construction that is shown in Figs. 222 and 223 is patented, and a royalty must be paid to the owners of the patent for each installation in which it is incorporated. Other types of warm-air panel construction are being developed, and an all steel house designed to incorporate warm-air panel heating may soon be produced on a mass production basis.

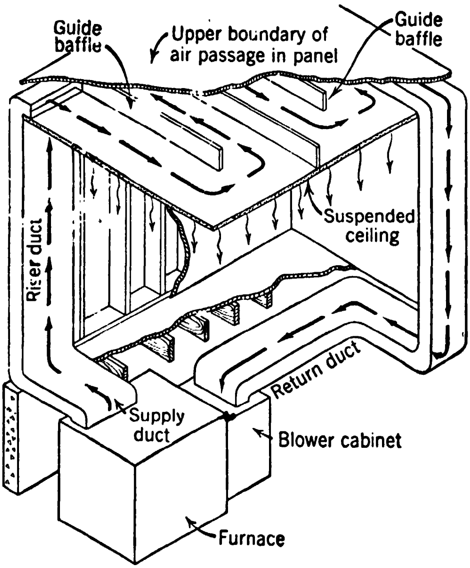


FIG. 223. Schematic diagram of a warm-air panel-heating system.

Where joists are the open-web steel type the lath and plaster or suitable substitute may be attached to the bottoms of the joists, as the open construction permits circulation of the air through the joist spaces. Figure 224 shows the construction of such a panel in the new Warm Air Furnace Research Residence located at the University of Illinois in Urbana, Illinois. The house is a one-story structure, and the view shown reveals the interior of the ceiling panel intended for the heating of the north bedroom. The completed construction included a layer of fireproof material laid over the top of the steel joists on which rock wool bats 4 in. thick were placed to reduce heat loss from the reverse side of the panel into the attic space. A layout of the five ceiling panels for this house is given in Fig. 225. The warm air supply ducts pass through the attic space from a plenum chamber above the attic floor to the panel entrance points always located near an outside wall. The path of the air flow through each panel is indicated by the arrows. The return air ducts which also

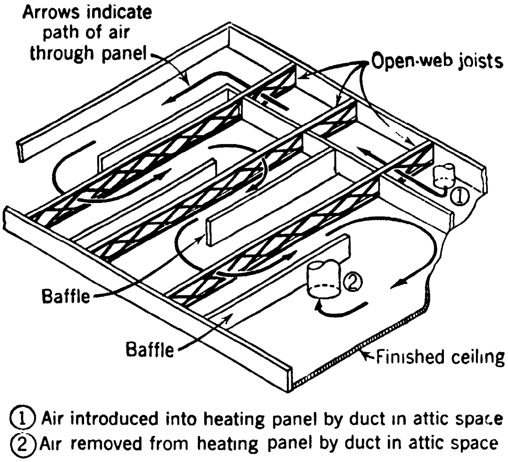
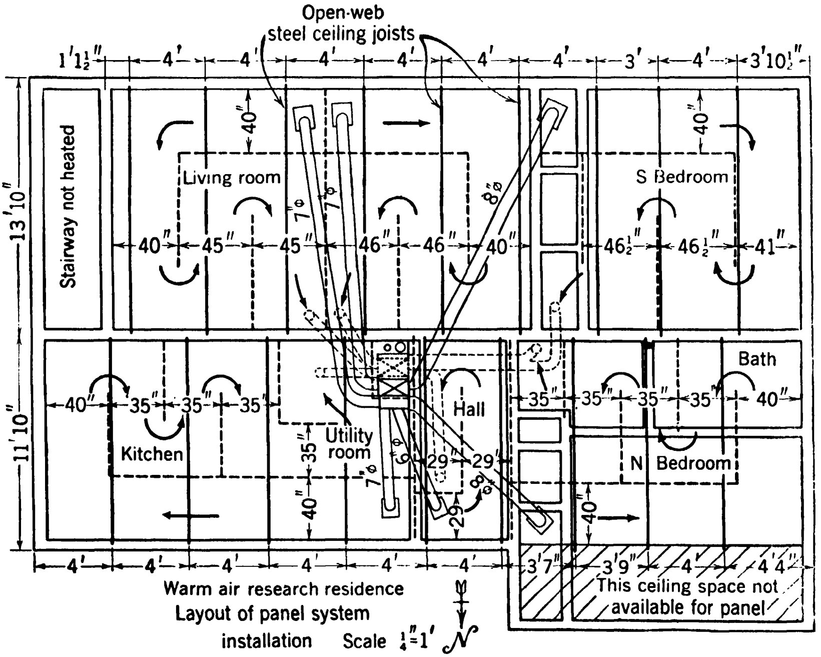

Arrows indicate path of air through panel
Open-web joists
①
②
Baffle
Baffle
Finished ceiling
① Air introduced into heating panel by duct in attic space
② Air removed from heating panel by duct in attic space

FIG. 224. A warm-air panel in which the air is circulated laterally through open-web steel joists. (Floor above joist space is not shown.)



(*National Warm Air Heating and Air Conditioning Research Residence, University of Illinois.*)

FIG. 225. Layout of warm-air ceiling panels. All ducts shown are in the attic space and are insulated with a blanket type of insulation. Ducts shown in solid lines supply heated air to the panels. Ducts shown in broken lines conduct the return air from the panels to the return-air plenum.

pass through the attic space receive the air from the panels and conduct it to the return-air plenum as indicated by the broken lines. Both the warm-air plenum and the return-air plenum are heavily insulated as are each of the two sets of air ducts. The warm-air plenum in the attic space is connected to the outlet from the furnace located in the basement by a vertical duct measuring 13 by 18 in. A similar vertical duct adjacent to the one first mentioned conducts the air from the return-air plenum also in the attic space to the lower part of the furnace casing. Rock-wool insulation of 4-in. thickness is provided between the two vertical ducts to prevent exchange of heat between the two air streams.

272. Design of Warm-Air Panel-Heating Systems. A suggested procedure to be used in designing a panel system, using heated air, is exactly the same as that outlined in Art. 217 through step 6. Beyond step 6 it is necessary to determine the required mean air temperature for the panel of greatest demand by solving for t_a in the equation, $H = UA(t_a - t_p)$, in which

$$t_a = \frac{H + UAt_p}{UA} \tag{117}$$

where H = estimated heat losses from the room, Btu per hr.

U = overall coefficient of heat transfer, heating air to panel surface, Btu per hr per sq ft per deg F.

A = area of the panel, sq ft.

t_a = average temperature of the heating air in the panel, deg F.

t_p = the required panel-surface temperature (from step 5, Art. 270), deg F.

After the required mean air temperature has been determined for the panel of greatest demand, the required areas of the remaining panels can be determined by the use of the same equation, in this case using the value of t_a found for the panel of greatest demand and solving for A. Often, as in the layout shown by Fig. 225, it may be more practicable to use the entire ceiling area as the panel in each room, though the required surface is less than this amount. When the panel areas are not adjusted to the heating requirements of the individual rooms the system must be balanced by regulating the volume of heated air that is fed into each panel. When the heating effect in the various rooms is balanced in this manner the temperature of the air recirculated from the different panels will not be the same and the air leaving the one which contains the greatest amount of excess area will be at the lowest temperature. Low temperature of the air leaving a panel means that its surface temperature is low near the exit point, which may result in uneven heating of the room below. Comfort conditions can usually be maintained in a room heated by such an arrangement, provided that the air passage through the panel is laid out in such a way that the warmest air circulates over the portions of the room which are close to outside walls, windows, or doors (see Fig. 225).

The weight of air to be circulated through each panel and through the system, assuming a 55 F drop in air temperature, is given by the equation

$$W = \frac{H}{0.24 \times 55} \tag{118}$$

where W = weight of air circulated, lb per hr.
H = the total heat to be supplied, including the losses from the reverse side of the panel, Btu per hr.

When the required weight of the air to be circulated has been determined the volume to be handled by the fan may be found by application of the following equation

$$Q = \frac{W}{60 \times d_a} \tag{119}$$

where Q = volume of air passing through the fan, cfm.
W = weight of air circulated, lb per hr.
d_a = the density of the air at the assumed barometric pressure and temperature at which it is returned from the panel, lb per cu ft.

The size of the distributing ducts and the friction loss of the system may be calculated by the method outlined in Art 302. A design procedure applicable to a system employing air-heated ceiling panels which is similar to that given in Chap. 6 for forced warm-air convection systems is included in Manual 7A,[1] published by The National Warm Air Heating and Air Conditioning Association.

273. Comparison of Warm-Air Panel Systems with Forced Circulation Convection Systems. The principle differences between a warm-air panel system and a conventional forced-circulation warm-air system are: (1) the panel system is a closed system in which the circulating air at no time enters the rooms which are heated, and (2) the allowed temperature drop of the circulating air is of the order of

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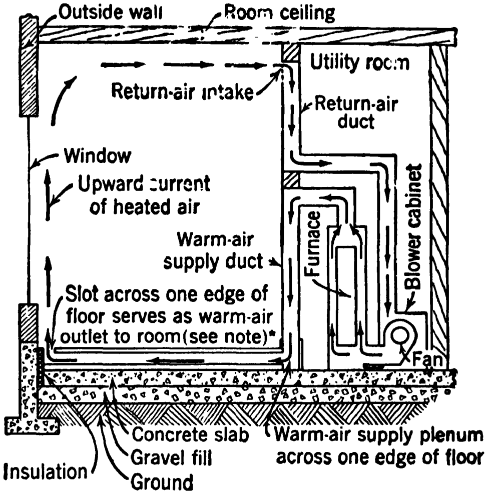
[1] "Code and Manual for the Design and Installation of Warm Air Ceiling Panel Systems," Copyright 1948 by National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland 14, Ohio.

55 F under design conditions compared with 100 F for the convection system. This means that a considerably greater weight of air will have to be circulated in the panel system, and the cost of power may be expected to be proportionately higher. Greater power costs in panel systems may be at least partially offset by improved heat transfer through the heating surfaces to the air passing through the furnace casing. Additional friction loss in the system may be offset by a reduction of friction loss through the elimination of the filter which is no longer needed. An important advantage of the panel system from the standpoint of comfort is that the temperature of floors or floor coverings is increased by the radiant heat received from the panel, regardless of the room surface in which it is placed. However, a heating panel in a room does not prevent downdrafts of cold air in areas close to windows and doors, whereas properly placed warm-air outlets in forced-circulation convection systems can be very effective in eliminating discomfort from this source. Another disadvantage of the panel system is that it does not afford a means of adding moisture to the air in the spaces served through evaporation of water in the furnace casing.



*Floor panel constructed of special hollow tile laid so as to form parallel air passages between supply plenum and outlet slot

FIG. 226. Combination forced-warm-air floor-panel and convection heating system.

274. Warm-Air Systems Combining Panel and Convection Heating. The advantage of warmer floors, characteristic of panel-heated rooms, can be obtained together with air circulation, in the spaces served, and air humidification by employing a combination system as is illustrated in principle by Fig. 226. In the combination system illustrated the warm air issuing from the slot at the edge of the panel is directed upward in a plane adjacent to the outer wall containing the greatest proportion of window area. This stream of heated air would be very effective in preventing downdrafts of cold air in that vicinity. The full advantage of the combination system can best be used when the panel is located in the floor instead of in the ceiling. Discharging the air which leaves the panel into the room instead of returning it directly to the furnace should permit a demand for increased heat output to be satisfied with less time lag than that which would result when all of the heat is delivered through the surface of the panel. The combination system has not been developed commercially at the time of this writing, but it appears to offer definite possibilities for results which are superior to those attainable by either a convection system or a panel system.

CHAPTER 13

AIR CONVEYING AND DISTRIBUTION, FANS, DUCT DESIGN, AND DIFFUSION

275. The Production of Air Flow. The movement of air is promoted either by differences in density or by applied pressure. Within a closed room natural air motion is created by heated air, of reduced density, rising and cooler air, of greater density, flowing to take its place. The movement is due to the process of convection, and the velocities of flow are low.

Gravity flow in ducts results from the difference in the weights of two columns of air of equal height and cross-section but of unequal temperatures. The action is the same as in chimneys and gravity-flow hot-water systems. Forced circulation of air in ducts arises from the pressure difference created by a fan or other agency which supplies the necessary pressure head to create the air velocity attained and to overcome frictional and other resistances to flow.

276. Volumes and Weights of Air Handled. The volume of a given weight of air varies as its temperature and pressure. Because of this fact, wherever possible it is desirable to convert air volumes to weight, by the use of the fundamental equation $PV = WRT$, before making calculations involving either heating or cooling of air. After the heat added or abstracted has been taken into account the air volume under the new conditions can be found.

Simple relationships hold for air quantities when Q is the volume, cubic feet per unit of time; W the weight, pounds per unit of time; d the weight, pounds per cubic foot; V the average velocity of flow, feet per unit of time; and A the cross-sectional area, of the duct in which flow occurs, square feet. The following are true: $Q = AV$, $A = Q/V$, $V = Q/A$, and $W = Qd$.

277. The Determination of Velocities of Air Flow. The instruments used to determine velocities of air flow in heating and air-conditioning systems depend upon their convenience of use and the magnitude of the velocity to be ascertained. Instruments employed are: kata thermometers, pitot tubes, and anemometers.

278. Kata Thermometer. The dry-bulb kata thermometer, Fig. 227, has an alcohol-filled bulb $\frac{1}{2}$ in. in diameter, and $\frac{5}{8}$ in. long, together with an 8-in. stem having engraved upon it graduations of 95 and 100 F. The instrument is used with air less than 90 F in temperature by immersing the bulb in warm water until alcohol is driven into the reservoir above the stem graduations. After careful drying of the bulb the thermometer is held away from the body, and the time in seconds for the alcohol to fall from 100 to 95 F is ascertained with a stop watch. The time required is a measure of the cooling power due to radiation and convection heat losses from the bulb. The readings vary with the dry-bulb temperature of the air and its velocity flow over the bulb. When the room-air temperature is above 100 F the bulb is cooled and dried and the time for the liquid rise from 95 to 100 F is determined.

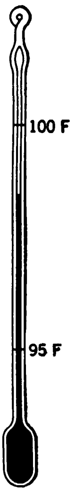


FIG. 227. Dry-bulb kata thermometer.

Each thermometer has a factor F marked on the stem. The factor F divided by either the average cooling or warming time, s, in seconds gives the cooling or heating power $K_p = F \div s$ in millicalories per second per square centimeter. Air velocities v, in feet per second, are calculated from the dry-bulb data by equation 120 for velocities more than 3.28 and equation 121 when the velocities are less than 3.28.

$$v = 3.28\left(\frac{K_p - 0.13t_c}{0.47t_c}\right)^2 \tag{120}$$

$$v = 3.28\left(\frac{K_p - 0.2t_c}{0.4t_c}\right)^2 \tag{121}$$

where t_c = 36.5 minus the dry-bulb temperature of the room air, C.

Dry-bulb kata thermometers are of importance only in the measurement of low velocities of air flow in occupied spaces.

279. Air-Duct Pressures. A discussion of the pressures existent within air ducts is desirable before the uses of pitot tubes and anemometers are considered. The *total pressure* is made up of two components, velocity pressure and static pressure.

Velocity pressure is utilized in creating the velocity of flow. *Static pressure* is that which tends to burst or collapse the duct and is used to overcome frictional and other resistances to air flow. Velocity, static, and total pressures are interrelated. When an increase of velocity of flow is produced at any part of a duct system, part of the static pressure at that point is required to produce the change of the rate of flow. Likewise, if the speed of flow is reduced at some point in a duct, part of the velocity pressure at that point is converted into static pressure. This phenomena is known as static-pressure regain. If the speed of flow is reduced gradually by the provision of a tapered transition section having a total included angle which does not exceed 7 deg, the static pressure increases by an amount that is only slightly less than the decrease in the velocity pressure.

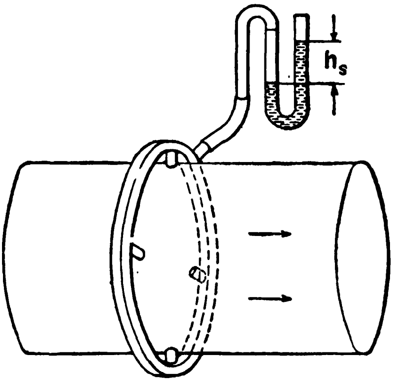


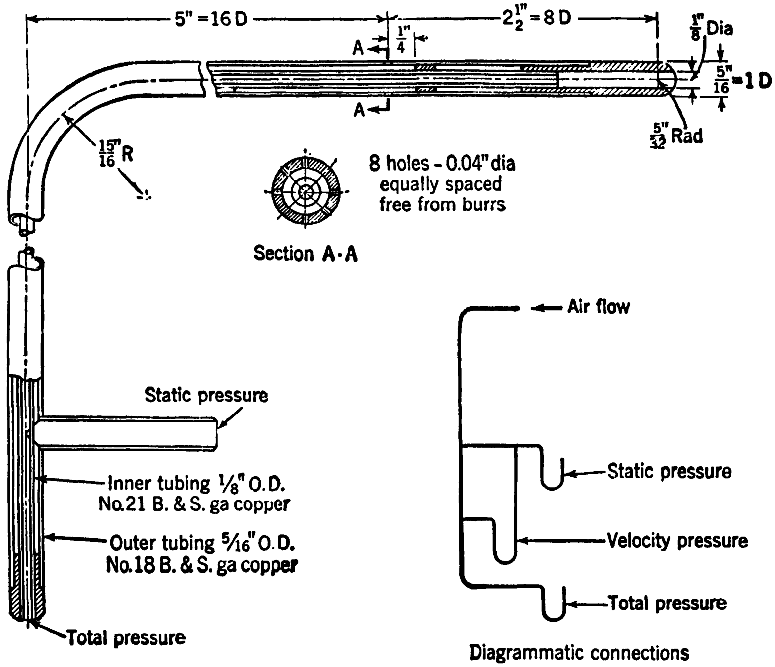
FIG. 228. Piezometer ring.

The static pressure existing within a duct may be determined by attaching a manometer or draft gage of sufficient capacity to connections made at right angles to the longitudinal axis of the duct, as in Fig. 228. Because of turbulence of air flow in ducts the piezometer ring connection is likely to give a truer indication of static pressure than one single connection. Velocity pressure is determined by the measurement of the total pressure and the subtraction of the static pressure from it, either mathematically or by the proper attachment of the static and total pressure connections to the indicating gage.

280. Pitot Tubes. A commercial type of pitot tube as adopted by the American Society of Heating and Ventilating Engineers, Fig. 229, combines in a single instrument both static- and total-pressure openings. A pitot tube for the measurement of total pressure only may be constructed of tubing of small internal bore and thin walls by bending a short leg at right angles to the remainder of the tube. The end of the short leg is cut squarely across and the tube walls ground to a thin edge at its end.

Either tube described is inserted in a duct with the short leg parallel to the longitudinal axis of the duct and with the open end opposing the air stream. The open end of the short leg allows the total of the impact of the air and its static pressure to be indicated by the gage attached.

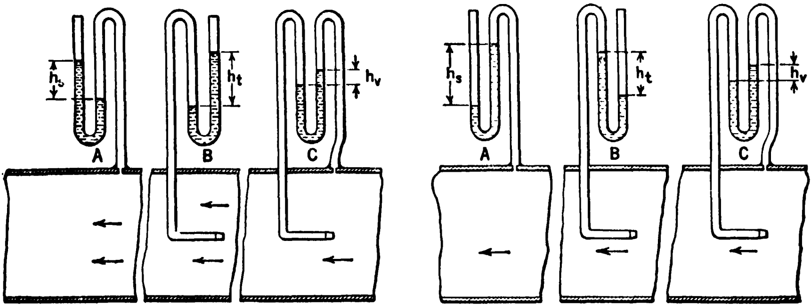
Pitot tubes are not adapted to the measurement of velocity pressures corresponding to flows less than 6 fps unless used with very sensitive and delicate pressure-measuring gages. Pressure measurements are best made in a straight section of pipe at least 20 pipe diameters in length with at least 10 duct diameters on either side of the tube loca-



Diagrammatic connections
Duct pressure above
atmospheric

FIG. 229. Pitot tube.

tion. Erroneous and variable data are likely to be secured when observations are taken near either a fan outlet or a bend in the duct.



Pressure greater than atmospheric

Pressure less than atmospheric

FIG. 230. Air-pressure measurements.

281. Pressures Greater and Less Than Atmospheric. Figure 230 indicates the use of a pitot tube in ducts where the air pressure is either greater or less than that of the outside atmosphere. The static pressure is represented by h_s, the total pressure by h_t, and the velocity pressure by h_v. The velocity pressure is always positive as measured. The static and total pressures are positive when the air in a duct is greater in pressure than the outside air and negative when the reverse is true, and the readings should be so indicated on the data sheet.

The velocity of flow in a duct is never uniform so that a number of observations are necessary in order to arrive at the average velocity of flow. A scheme of locating a tube at several points on two diameters of circular ducts and in rectangular and square ducts is shown by Fig. 231. Observations taken at the points indicated serve to determine the average velocity of flow.

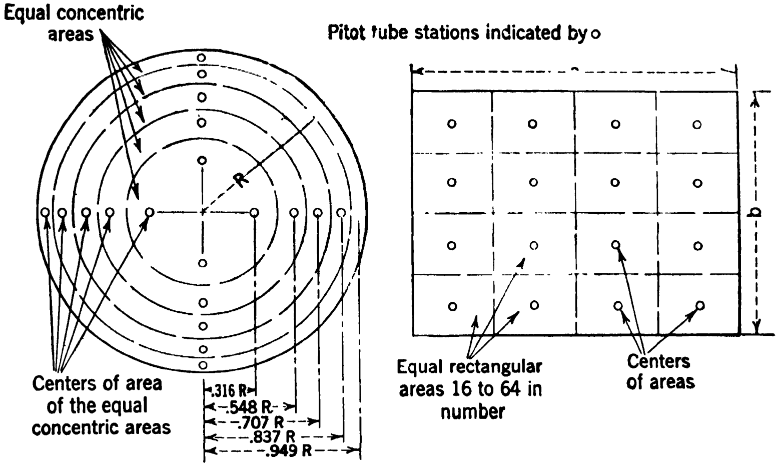

Equal concentric areas
Pitot tube stations indicated by o
R
D
Centers of area of the equal concentric areas
.316 R
.548 R
.707 R
.837 R
.949 R
Equal rectangular areas 16 to 64 in number
Centers of areas

FIG. 231. Locations of a pitot tube for a duct traverse.

282. Computation of Velocity of Flow from the Velocity Pressure. The velocity of air flow is readily calculated from measured velocity pressures by the equation

$$v = \sqrt{2g \frac{d_w h_v}{12 d_a}} \tag{122}$$

where v = velocity, fps.
g = acceleration due to gravity, 32.16 ft per sec per sec.
d_w = weight of 1 cubic foot of water at the temperature of the gage fluid, Table 4, Chap. 1.
h_v = velocity pressure measured, in. of water.
d_a = density of the air flowing, lb per cu ft.

The average velocity of flow in a duct represents the mean of all the velocities at various places in the section of a duct at any place. The

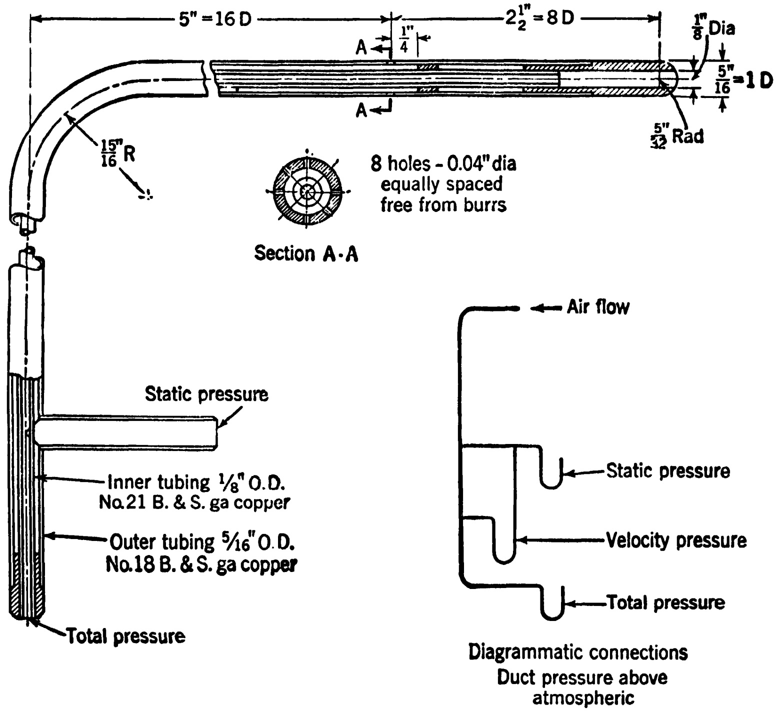
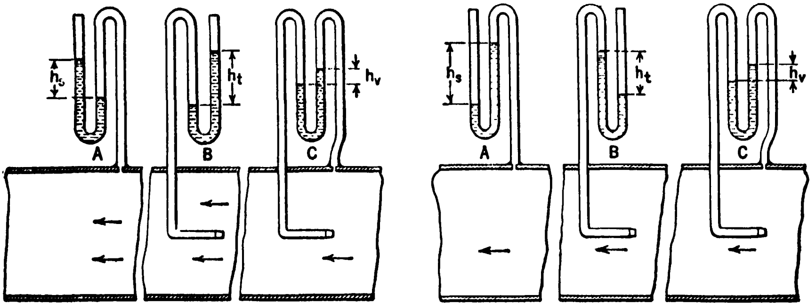


FIG. 229. Pitot tube.

tion. Erroneous and variable data are likely to be secured when observations are taken near either a fan outlet or a bend in the duct.



Pressure greater than atmospheric

Pressure less than atmospheric

FIG. 230. Air-pressure measurements.

281. Pressures Greater and Less Than Atmospheric. Figure 230 indicates the use of a pitot tube in ducts where the air pressure is either greater or less than that of the outside atmosphere. The static pressure is represented by h_s, the total pressure by h_t, and the velocity pressure by h_v. The velocity pressure is always positive as measured. The static and total pressures are positive when the air in a duct is greater in pressure than the outside air and negative when the reverse is true, and the readings should be so indicated on the data sheet.

The velocity of flow in a duct is never uniform so that a number of observations are necessary in order to arrive at the average velocity of flow. A scheme of locating a tube at several points on two diameters of circular ducts and in rectangular and square ducts is shown by Fig. 231. Observations taken at the points indicated serve to determine the average velocity of flow.

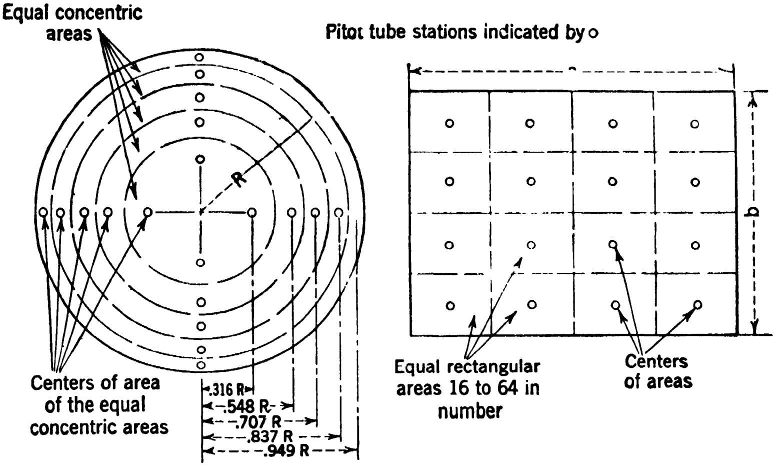

Equal concentric areas
Pitot tube stations indicated by o
R
Centers of area of the equal concentric areas
.316 R
.548 R
.707 R
.837 R
.949 R
D
Equal rectangular areas 16 to 64 in number
Centers of areas

FIG. 231. Locations of a pitot tube for a duct traverse.

282. Computation of Velocity of Flow from the Velocity Pressure. The velocity of air flow is readily calculated from measured velocity pressures by the equation

$$v = \sqrt{2g\frac{d_w h_v}{12 d_a}} \tag{122}$$

where v = velocity, fps.
g = acceleration due to gravity, 32.16 ft per sec per sec.
d_w = weight of 1 cubic foot of water at the temperature of the gage fluid, Table 4, Chap. 1.
h_v = velocity pressure measured, in. of water.
d_a = density of the air flowing, lb per cu ft.

The average velocity of flow in a duct represents the mean of all the velocities at various places in the section of a duct at any place. The square root of the velocity pressure is involved in the calculation of a velocity. Therefore in arriving at the average velocity of flow the average square root of the velocity pressures must be used and not the square root of the average of the velocity pressures read. Thus

$$\sqrt{h_{va}} = \frac{\sqrt{h_{v_1}} + \sqrt{h_{v_2}} + \sqrt{h_{v_3}} \cdots + \sqrt{h_{v_n}}}{n} \tag{123}$$

where n is the number of observations taken. For average velocity of flow v_a equation 122 should be arranged as

$$v_a = \sqrt{2g\,\frac{d_w}{12d_a}} \times \sqrt{h_{va}} \tag{124}$$

283. Anemometers. Figure 232 illustrates one form of an instrument known as an anemometer which is used for the determination of air velocities ranging from 150 to 1500 fpm. This type of anemometer is a delicate instrument which requires frequent calibration and careful handling. The air flow causes the vaned wheel to rotate and operate the indicating mechanism.

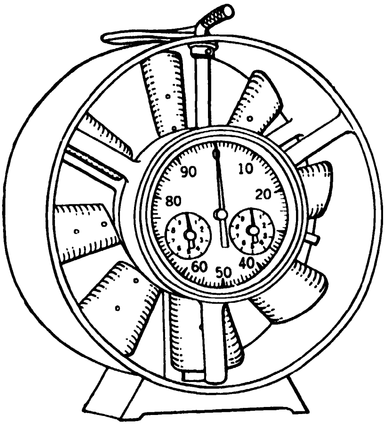


FIG. 232. Anemometer.

In order to secure the average velocity of flow it is necessary to traverse the area in question as in the use of a pitot tube, holding the anemometer for equal intervals of time at the different locations of the traverse. The difference of the final and the initial readings of the anemometer dial divided by the time required for the traverse gives the average velocity, in feet per unit of time involved, for their unrestricted inlets and outlets. When the traverse is made either at a duct air-intake opening or a duct air-outlet opening fitted with a grille, the volume of air flow is computed as indicated by equation 125.[1]

$$Q = \frac{CV(A + a)}{2} \tag{125}$$

---

[1] The Measurement of the Flow of Air through Registers and Grilles, by Lynn E. Davies, *ASHVE Trans.*, Vol. 36, 1930.

where Q = air volume, cfm.
C = constant from Table 91.
V = velocity of flow as indicated by the anemometer and corrected by reference to the calibration data, fpm.
A = gross area of the grille, sq ft.
a = free area of the grille openings, sq ft.

When the flow is inward through the grille into the duct, during a traverse, the anemometer is placed with the edge of the protecting ring about the wheel against the grille face. For outward flow, the anemometer is held at the different locations at a distance of 3 in. from the grille face. Values for the constant C are given in Table 91.

TABLE 91

VOLUME COEFFICIENTS C

Average Velocity of Flow Indicated by Anemometer, Fpm	Discharge Grilles	Intake Grilles	Average Velocity of Flow Indicated by Anemometer, Fpm	Discharge Grilles	Intake Grilles
150	0.952	0.993	500	0.985	1.067
200	0.957	1.005	600	0.992	1.078
300	0.967	1.028	700	0.998	1.084
400	0.977	1.049	800	1.000	...

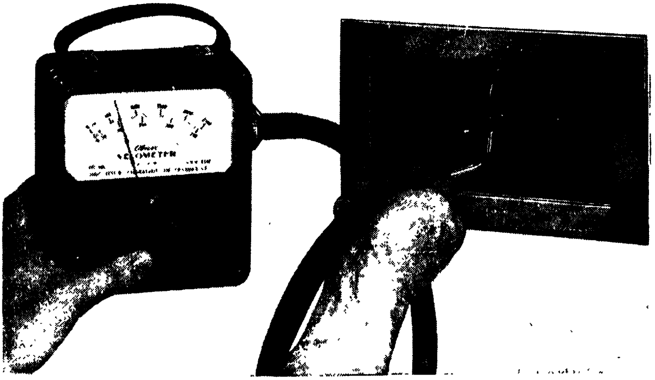
Whenever a velocity pressure, corresponding to a velocity of flow, is required, equation 122 is arranged as

$$h_v = \frac{12v^2 d_a}{2gd_w} \tag{126}$$

to give the velocity pressure. The units are the same as in equation 122.

A more recently developed instrument known as a vane anemometer or a velometer is valuable because it gives an instantaneous reading. This device, one form of which is illustrated in Fig. 233, consists of a delicately balanced and magnetically stabilized vane within a case through which the air flows when it is pointed into the air stream. Velocities as low as 50 fpm may be measured, and attachments may be provided to adapt the instrument to the measurement of the flow of air in ducts.

The hot-wire anemometer is still another instrument adapted to the measurement of air velocities ranging from 20 to 2000 fpm. In this instrument a constant current is maintained through a wire, and the observed temperature of the wire is a measure of the air movement when referred to the proper calibration data. An advantage of this instrument is that it can be arranged so as to be free from directional effect.



(*Illinois Testing Laboratories, Inc.*)

FIG. 233. Determination of return-grille-face air velocity with a velometer. A change of tube tips permits the measurement of supply-grille air velocities.

284. Fans. Air movement through heating, ventilating, and air-conditioning apparatus is usually accomplished through the use of a fan. In past years the centrifugal fan in several different designs has been used almost exclusively, but, because of certain advantages which will be discussed later, the axial-flow type of fan is now gaining favor with the air-conditioning engineer.

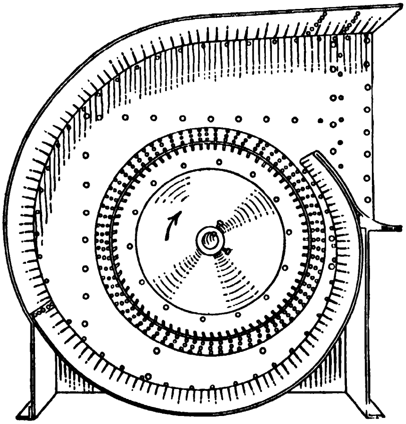


FIG. 234. Fan wheel and scroll.

285. Centrifugal Fans. Centrifugal fans are made in two types, namely, (1) steel plate and (2) multiblade. Either type or both types may be used where positive pressures are necessary and where gravity flow of air is not feasible.

All centrifugal fans consist of a wheel which is rotated within a housing usually constructed of sheet metal. The housing of steel-plate and multivane fans is of a scroll shape, as in Fig. 234, that permits the air or gas to be carried away from the periphery of the wheel with slight disturbances and power losses.

Fan wheels have some sort of vanes or blades located at or near the wheel rim. The action produced by a wheel in rotation is secured as a result of the tendency of air, adjacent to the forward sides of the fan blades, to flow radially outward. This action is due to centrifugal force, and the air is discharged from the tips of the blades into the scroll of the housing. As a result of this movement a pressure less than atmosphere is produced at the center of the wheel, and a positive air pressure is built up in the scroll. Air or gas flows axially into the wheel to take the place of that discharged. Consequently, not only may centrifugal fans be used to exhaust air or gases through a duct system attached to the fan-housing inlet, but also they may be made to discharge the same air or gases under several inches of water pressure through a duct system leading away from the housing scroll.

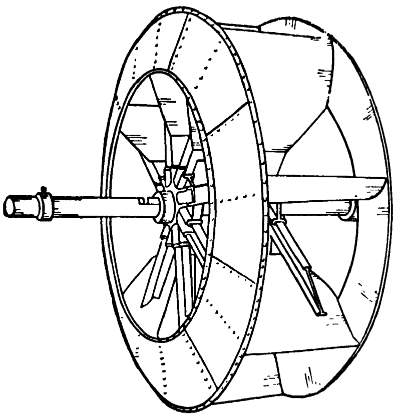


Fig. 235. Steel-plate fan wheel.

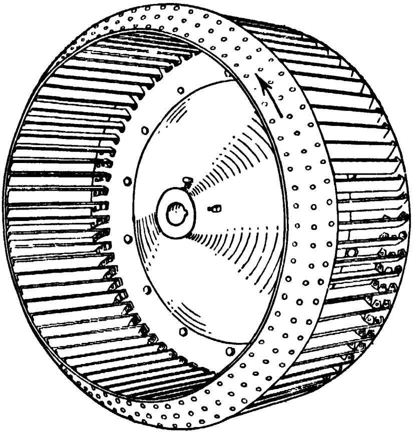


Fig. 236. Forward-curved-blade fan wheel.

286. Steel-Plate Fans. In fans of this type the wheels consist of one or two spiders, each having from 6 to 12 arms. Each pair of arms carries a flat steel float or blade of some radial length as in the typical wheel of Fig. 235. The wheel floats may be straight or curved either forward or backward, depending upon the operating characteristics desired. The blade curvature has a marked effect upon the performance of a given type of fan.

287. Multiblade Fans. Such fan wheels are built up of two or more annular rings with many narrow curved blades inserted between them as illustrated by Figs. 236 and 237. The blades may be curved either forward, Fig. 236, or backward, Fig. 237, depending upon the performance characteristics desired. Multiblade wheels designed for high speeds and pressures are reinforced by a number of annular rings, and the blades are made short laterally, in order to secure greater rigidity.

Fans having wheels as shown by Fig. 238 are of the multiblade type, and the term conoidal is applied to the blades which are in the form of the surfaces of two tangent cones. This design gives the blade a forward curve at its inward edge and a marked backward curvature at the periphery or exit edge. A blade producing desirable operating characteristics is secured which also has sufficient rigidity without the use of stiffening rings.

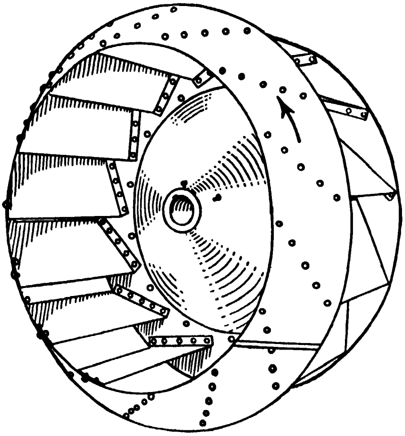


FIG. 237. Backward-curved-blade fan wheel.

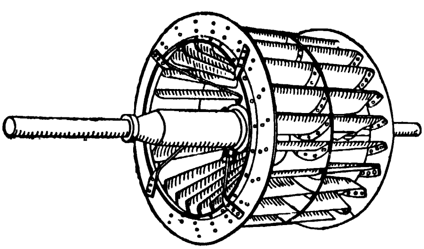
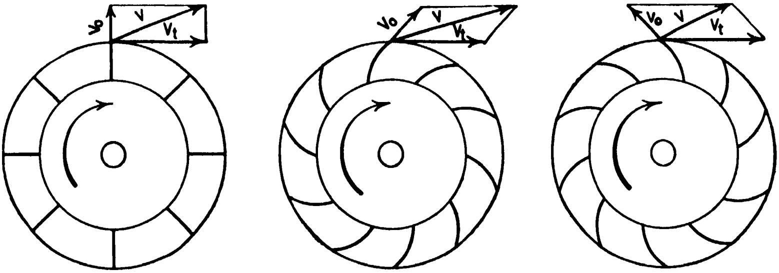


FIG. 238. Conoidal double-inlet, double-width fan wheel.



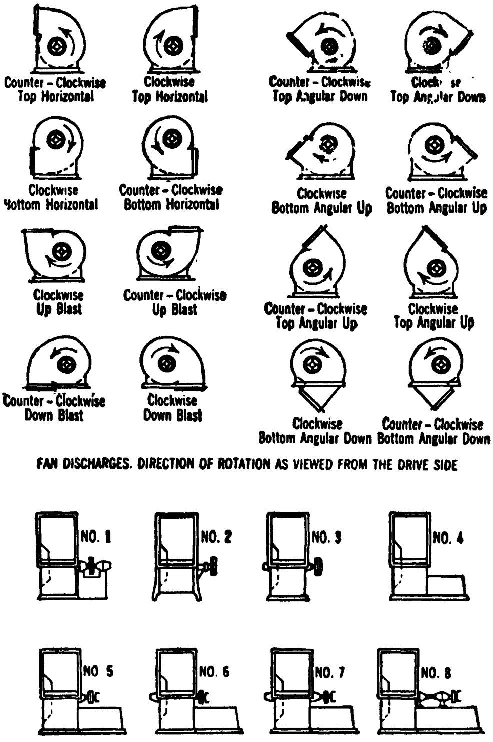
Straight Blade

Forward Curved Blade

Backward Curved Blade

FIG. 239. Types of fan blades.

288. Effects of Fan-Blade Shapes on Air Velocities. The effects of blade shapes upon the resultant velocity of air are indicated in Fig. 239. The vector diagrams indicate velocities by the following symbols: blade-tip velocity V_t, velocity of the air flowing along the blade face V_o, and the final resultant velocity of the air V. When all the wheels shown are operated with the same blade-tip speed the forward-curved blades give the highest and the backward-curved blades the smallest resultant air velocity. The resultant velocity of the air as it leaves the fan blades is of importance from the standpoint of operating speed and the noise which may be produced by the fan. Undue noise cannot be tolerated in many heating and air-conditioning systems.


Counter-Clockwise Top Horizontal
Clockwise Top Horizontal
Counter-Clockwise Top Angular Down
Clockwise Top Angular Down
Clockwise Bottom Horizontal
Counter-Clockwise Bottom Horizontal
Clockwise Bottom Angular Up
Counter-Clockwise Bottom Angular Up
Clockwise Up Blast
Counter-Clockwise Up Blast
Counter-Clockwise Top Angular Up
Clockwise Top Angular Up
Counter-Clockwise Down Blast
Clockwise Down Blast
Clockwise Bottom Angular Down
Counter-Clockwise Bottom Angular Down
FAN DISCHARGES. DIRECTION OF ROTATION AS VIEWED FROM THE DRIVE SIDE
NO. 1
NO. 2
NO. 3
NO. 4
NO 5
NO. 6
NO. 7
NO. 8

ARRANGEMENT OF DRIVE

FOR BELT DRIVE

Arr. 1. Single-width fan with either overhung wheel or overhung pulley and two bearings.

Arr. 2: Single-width fan with bracket bearing for overhung wheel and overhung pulley.

Arr. 3: Single-width fan with overhung pulley. Two bearings.

FOR DIRECT-CONNECTED DRIVE

Arr. 4: Single-width fan with extended base. Wheel overhung on engine or motor shaft.

Arr. 5: Single-width fan with extended base and coupling. Bearing on the drive side.

Arr. 6: Single-width fan with extended base and coupling. Bearing on the inlet side.

Arr. 7: Single-width fan with extended base, two bearings, and coupling.

Arr. 8: Single-width fan with extended base and coupling. Similar to Arr. 5 except two bearings on pedestal for motor or drive and furnished with flexible coupling.

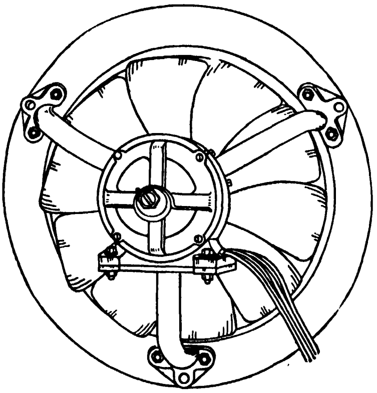
FIG. 240. Fan discharges and arrangements of drive.

Backward-curved blades in fans permit high operating speeds without excessive noise and produce other desirable operating characteristics, such as high volumetric efficiencies and a wide range of capacity at constant speed with small changes in the power requirements.

289. Fan Nomenclature. Fans of the steel-plate and multiblade types are designated according to (1) the number of inlets, single or double; (2) the width of the wheel, single or double; (3) the discharge, top, bottom, vertical, horizontal, or angular; (4) the housing, full, seven-eighths, or three-quarters; and (5) clockwise and counterclockwise rotation as viewed from the drive side.

A full-housed fan is one in which the fan scroll is completely above the base upon which the fan rests, as illustrated by Fig. 234. Seven-eighths or three-quarters full-housed fans have the scroll extending below the top of the supporting base. More of the scroll is located below the base line in the three-quarters full-housed than in the seven-eighths full-housed fan. The standardized designations of the hand of a centrifugal fan and drive arrangements are shown in Fig. 240.

290. Propeller and Disc Fans. Fans of these types have two or more blades mounted on the hub of the wheel, Fig. 241. The blades extend outward radially from the hub and are set at an angle with the longitudinal axis of the wheel shaft. Such fans are used with some unit heaters and coolers for forcing air over coils and moving it into the space served. Many propeller and disc fans will not handle air against much resistance. This fact, when such is the case, makes them unsuitable for systems having ducts, grilles, heaters, coolers, air filters, and air washers. Airplane-type propeller fan wheels can be used with some duct systems.



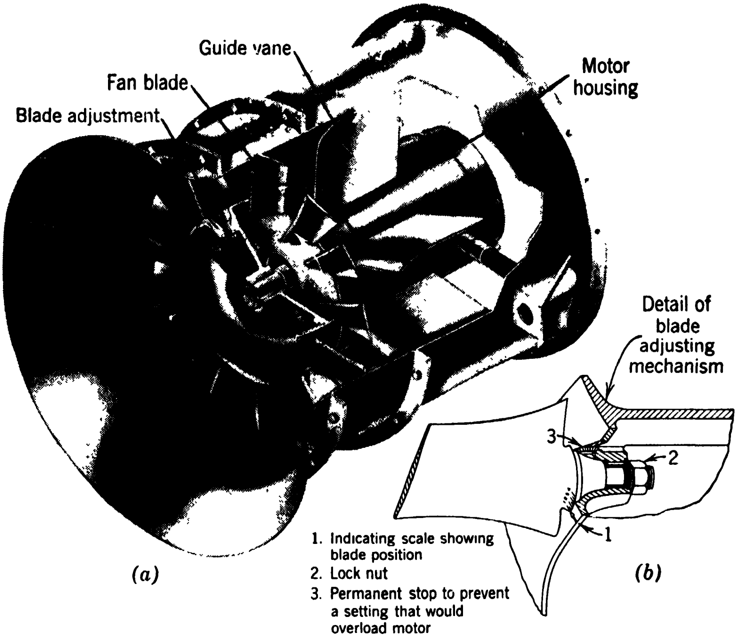
(*American Blower Corp.*)

FIG. 241. Disc fan.

All propeller and disc fans have a place in the removal of air from spaces when the fan unit can be placed in an opening in the wall, and discharge of the air is made to the outside without a duct. One objection to some disc and propeller fans is the noise they make at high speeds.

291. Axial-Flow Fans. In recent years certain fan manufacturers have developed a special type of propeller fan which is adapted to the movement of large volumes of air against considerable frictional resistance. Fans of this type have large hubs and short blades of special air-foil shape and are designated as "axial-flow" to distinguish them from the earlier propeller and disc types. Figure 242 shows an illustration of such a fan provided with stationary vanes in the outlet to prevent rotational flow of air in the discharge duct. The fan shown also includes a provision for resetting the blade angles after installation as a compensation for either miscalculation of the resistance to airflow or poorly installed duct work.

The axial-flow type of fan should be considered in selecting an air-propelling unit for an air-conditioning system, as certain types are now available which are capable of delivering any required volume of air against static pressures up to 9 in. of water while operating at efficiencies and noise levels that compare favorably with the corresponding characteristics of centrifugal fans. This type of fan, because its discharge opening is in line with its entrance, offers the advantage of a simplified duct arrangement in the fan room. This advantage is of especial importance where space is valuable or when a large fan is to be placed in a room having a low ceiling.


Guide vane
Fan blade
Blade adjustment
Motor housing
Detail of blade adjusting mechanism
3
2
1
1. Indicating scale showing blade position
2. Lock nut
3. Permanent stop to prevent a setting that would overload motor
(a)
(b)

(*Joy Manufacturing Co.*)

FIG. 242. Axial-flow fan with adjustable blades.

292. Total-Pressure Difference Developed by a Fan. Fans operate with the gases handled arriving at the inlet to the fan housing under atmospheric pressure when no inlet duct is used and at pressures less than atmospheric when exhausting through an inlet duct. Certain types of blowers such as gas boosters have pressures greater than atmospheric in the inlet duct to the blower or fan.

If the fan has no suction duct the entry losses to the fan housing are considered as part of the fan losses and are reflected in the mechanical efficiency of the fan. If the fan has no discharge duct leading away from its outlet the discharge static pressure is zero, and the total pressure at the discharge outlet is equal to the average velocity pressure at the fan outlet. In any event, the average total-pressure difference (total pressure) created by a fan is the average total pressure at the fan outlet minus the average total pressure at the fan inlet.

Example. A fan having no inlet duct discharged air into a duct at 1.00 in. of water static pressure and an average velocity pressure of 0.15 in. of water. Find the total pressure difference developed by the fan.

Solution.

$$h_T = [1.00 + 0.15] - 0$$

$$= 1.15 - 0 = 1.15 \text{ in. of water}$$

Example. A fan maintained at its outlet an average static pressure of 1.25 in. of water with an average outlet velocity pressure of 0.35 in. of water. In the suction duct near the fan inlet the static pressure was −1.25 in. of water and the velocity pressure was 0.25 in. of water. Find the total-pressure difference developed by the fan.

Solution.

$$h_T = [1.25 + 0.35] - [-1.25 + 0.25]$$

$$= 1.60 + 1.00 = 2.60 \text{ in. of water}$$

Example. A blower received gas at +0.25 in. of water static pressure and a velocity pressure of 0.35 in. of water in the inlet duct. The discharge static pressure was 15 in. of water with a velocity pressure of 0.75 in. of water. Find the total-pressure difference produced by the blower.

Solution.

$$h_T = [15.0 + 0.75] - [0.25 + 0.35]$$

$$= 15.75 - 0.60 = 15.15 \text{ in. of water}$$

293. Fan-Air Discharge. The volume of gas flowing is equal to the product of the duct section in square feet and the average velocity of flow at that section.

$$Q = Av_a \qquad (127)$$

where Q = cu ft discharged per sec.
A = duct cross-section area, sq ft.
v_a = average velocity of flow, fps.

294. Fan-Air Horsepower. The power output of a fan is expressed in terms of air horsepower (ahp) and represents work done by the fan.

$$ahp = \frac{W_aH}{33{,}000} \tag{128}$$

where W_a = weight of air handled per minute, lb.
H = total head against which the fan works, ft of air.

In terms of the volume of air handled and the total pressure difference created by the fan,

$$ahp = \frac{Q \times 60 \times h_T \times d_w}{12 \times 33{,}000} \tag{129}$$

where Q = volume of air handled, cfs
h_T = total pressure difference created by the fan, in. of water.
d_w = density of one cubic foot of water at the temperature of the gage fluid, lb.

Example. A fan, having a free inlet, discharges 24,000 cfm through its outlet of 12.6 sq ft area and maintains a static pressure of 5 in. of water. The air temperature is 70 F, and the barometric pressure 29.92 in. of mercury. Find the air horsepower developed.

Solution.

$$v_a = \frac{24{,}000}{12.6 \times 60} = 31.8 \text{ fps}$$

$$h_{va} = \frac{12v_a{}^2d}{2gD} = \frac{12 \times 31.8^2 \times 0.075}{2 \times 32.16 \times 62.3} = 0.226 \text{ in. of water}$$

$$h_T = h_s + h_{va} = 5. + 0.226 = 5.226 \text{ in. of water}$$

$$ahp = \frac{24{,}000 \times 5.226 \times 62.3}{12 \times 33{,}000} = 19.8$$

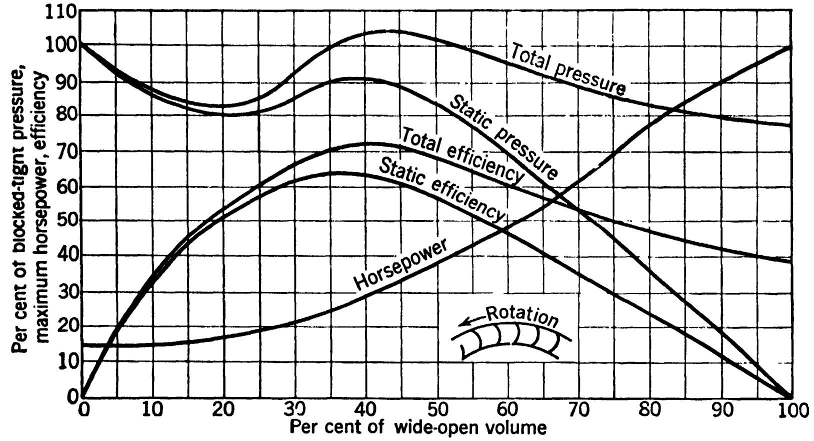
295. Fan Efficiencies. The ratio of the air horsepower output to the driving power (brake horsepower) required at the fan shaft expresses the mechanical efficiency of the fan.

$$\text{Fan mechanical efficiency,} \quad e_m = \frac{\text{Air horsepower}}{\text{Horsepower input}} \times 100 \tag{130}$$

The air horsepower is a function of the static pressure developed by the fan and of the velocity pressure at the fan discharge. The velocity pressure is, however, not entirely useful in overcoming frictional resistance, as some of it is converted to heat because of turbulence, and some of it is discharged from the system as kinetic energy in the leaving air streams. Therefore a method of calculating the efficiency based on the static pressure only is often used in reporting fan performance. This static efficiency e_s is the mechanical efficiency e_m multiplied by the ratio of the static pressure to the total pressure.

$$e_s = e_m \times \frac{h_s}{h_t} \qquad (131)$$

The mechanical efficiencies of steel-plate fans range from 40 to 60 per cent while those of multiblade-centrifugal or axial-flow fans range from 50 to 90 per cent. Disc- and propeller-type fans operate at efficiencies approaching 100 per cent when the resistance to air flow is small, but the efficiency decreases rapidly as the resistance is increased. Inasmuch as the mechanical efficiency as given by equation 130 is based upon the air horsepower which in turn is based upon the total pressure, this performance characteristic is often referred to as the total efficiency to distinguish it from the static efficiency.



(*From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.*)

FIG. 243. Operating characteristics of a fan with blades curved forward.

296. Fan Performance. Fans should be tested according to the Standard Test Code for Centrifugal and Axial Fans adopted by both the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers. The conditions under which the capacities are to be given involve standard air weighing 0.075 lb per cu ft.

Centrifugal and axial fans are arranged for blowing tests by setting them up without a duct connection at the inlet and with a discharge duct having a uniform cross-sectional area equal to that of the fan discharge outlet and not less than 10 duct diameters in length. When a transformation piece is used at the fan outlet to change the shape of the section from rectangular to round the greatest angle between the longitudinal axis of the duct and any element in the sides of the adapter is not to be more than 7 deg.

Pressure measurements are made at a point not less than three-fourths of the duct length away from the fan outlet. The duct outlet is arranged so that different amounts of opening ranging from 0 to 100 per cent may be had. Fans under test are first operated at constant speed and data taken with the percentages of the duct opening ranging from 0 to 100 per cent. The taking of such data permits the development of characteristic curves similar to those of Figs. 243, 244, and 245. The maximum efficiency of a fan occurs when operated at constant speed at some duct opening between 0 and 100 per cent.

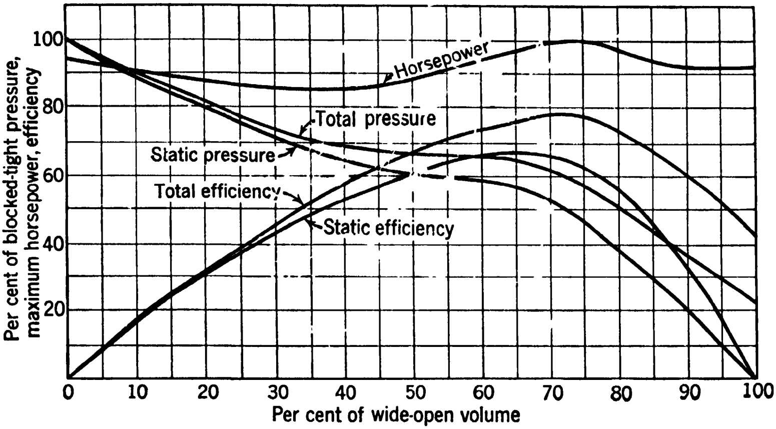


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FIG. 244. Operating characteristics of axial-flow airfoil-type fan.

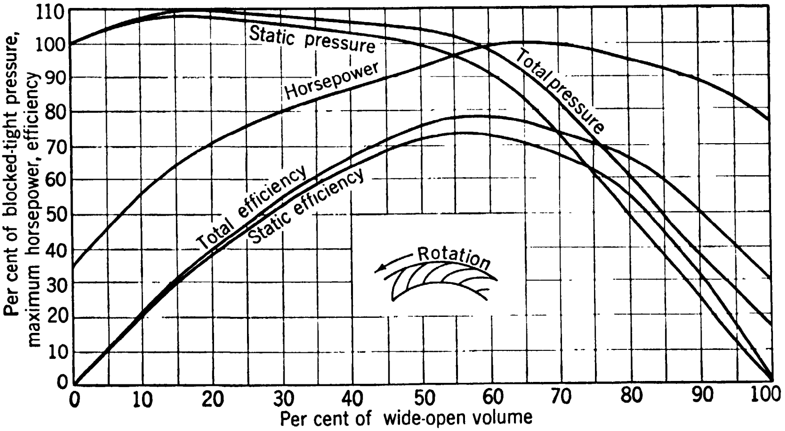

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FIG. 245. Operating characteristics of a fan with blades curved backward.

The several manufacturers of fans have different methods of plotting the characteristic curves and care must be taken to ascertain how the material is presented. Thus the characteristic curves may be plotted, as shown in the above-mentioned figures, with the percentage of wide-open volume as abscissa and the percentages of blocked-tight pressures, maximum brake horsepower, and efficiencies as ordinates. Again, the curves may be the same as the foregoing except that the percentages of the duct opening are used as abscissas. Percentages of air flow with 100 per cent duct opening and percentages of duct opening are not the same, as the flow of air in a duct with a fixed fan speed and 50 per cent duct opening is not one-half the flow at 100 per cent duct opening.

It may be noted from Fig. 245 that centrifugal fans having backward-curved blades require maximum brake horsepower when the volume handled is intermediate between zero and that for wide-open conditions. It may also be noted from Fig. 245 that the operating condition which requires the maximum brake horsepower is close to the combination of volume and static pressure under which the fan operates most efficiently. Fans of this type are said to have a non-overloading power characteristic which means that the driving motor cannot be overloaded if the fan and motor are properly selected. Axial-flow fans also have non-overloading power characteristics (see Fig. 244), but centrifugal fans whose blades are tipped forward require an ever-increasing amount of power as the volume is increased from zero to that flowing under wide-open conditions (see Fig. 243). However, this type of centrifugal fan provides greater static pressure for a given blade-tip velocity than the other types which have been discussed and is commonly used in air-conditioning systems in spite of the disadvantage which has been mentioned. Changes in a system which might reduce its overall resistance to flow should be made with caution when a fan of this type is included, as such a change may result in a serious overload on the driving motor.

297. Effect of System Resistance on the Volume Delivered by a Fan. When a fan is in actual operation as part of an air-handling system the static pressure it develops must be the same as the resistance of the attached system. The resistance of any fixed system varies nearly as the square of the flow and may be represented as a curve in which the static pressure necessary to overcome it is plotted against the volume handled. Curve *A* of Fig. 246 may be assumed to show the resistance of a certain system for varying amounts of flow. Such a curve may be referred to as a system characteristic. Curve 1 may be assumed to show the static pressure which a certain fan is capable of developing with the same variation in air volume when operated at a certain speed.

It may be assumed that the fan whose static pressure characteristic at a certain speed is curve 1 becomes part of a system whose resistance characteristic is curve *A*. When the fan is put into operation the resistance of the system will increase with increase in air volume, along curve *A*, until this curve intersects curve 1. At this point the resistance of the system is equal to the static pressure which the fan is capable of producing, and the flow becomes stabilized at the amount *X* on the volume scale directly below the intersection. If the resistance of the system should be increased by the partial clogging of a filter, the system characteristic might be changed from curve *A* to curve *B* in which case the volume delivered would be reduced from amount *X* to amount *Y*. It would be possible to maintain volume *X* against the increased resistance of the system represented by curve *B* by increasing the fan speed so as to change the fan static pressure characteristic from curve 1 to curve 2, provided that the motor were capable of delivering the increased power requirement. At the increased speed this fan would handle the volume represented by the point *Z* if the filter were cleaned or replaced so as to return the system resistance to that represented by curve *A*.

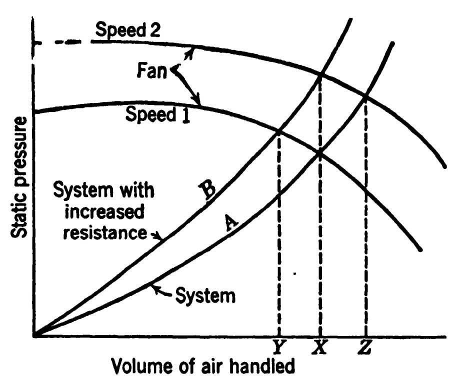


Fig. 246. Operating characteristics of a fan and duct system.

298. Fan Tables. Fans are also operated at a constant static pressure at variable speeds to give several capacities with differing power requirements. The usual tabular data relative to the performance of a given size of a centrifugal fan, as indicated by the builder, are similar to those of Table 92. The operating data for maximum efficiency are usually designated in some way, either by bold-faced type, italics, or underscoring.

Table 93 gives similar data in abridged form for nine different-sized fans of the same type. The data in Table 93 were selected from tables of the form of Table 92 and show, for each fan and each static pressure, only the volume of air which can be delivered most efficiently. The outlet velocity, the required fan speed, and the required brake horse-

TABLE 92

CAPACITIES OF A NUMBER 2 TYPE-HV FAN WITH FORWARD-CURVED BLADES*

Single inlet, wheel diameter 26 in., wheel circumference 6.81 ft. Air density 0.075 lb per cu ft. Single width, size of outlet $19\frac{3}{4}$ by $25\frac{5}{8}$ in.

Volume,	Outlet Velocity,	¼-In. Static Pressure,		½-In. Static Pressure,		¾-In. Static Pressure,		1-In. Static Pressure,		1¼-In. Static Pressure,		1½-In. Static Pressure,		1¾-In. Static Pressure,		2-In. Static Pressure ,		2½-In. Static Pressure,		3-In. Static Pressure,	
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
3,440	1000	*226*†	*0.26*																		
3,784	1100	235	0.32	*294*	*0.50*	*356*	*0.68*														
4,128	1200	245	0.38	301	0.56	358	0.76	411	0.97												
4,472	1300	255	0.47	308	0.64	363	0.85	*411*	*1.04*	460	1.31										
4,816	1400	268	0.56	319	0.71	367	0.95	411	1.18	460	1.42	507	1.66								
5,160	1500	279	0.64	325	0.83	374	1.04	417	1.28	*460*	*1.52*	507	1.80	547	2.14						
5,504	1600	291	0.73	338	0.95	382	1.18	418	1.38	463	1.66	*507*	*2.00*	544	2.28	584	2.59				
5,842	1700	304	0.88	348	1.07	389	1.31	426	1.56	463	1.80	507	2.18	*544*	*2.42*	580	2.73	650	3.38		
6,192	1800	318	1.00	360	1.21	397	1.47	433	1.73	470	2.00	512	2.31	544	2.59	*580*	*2.93*	647	3.62	717	4.35
6,536	1900			371	1.38	408	1.64	444	1.93	475	2.18	517	2.52	551	2.76	580	3.07	647	3.83	713	4.63
6,880	2000			382	1.55	419	1.81	452	2.01	485	2.35	524	2.69	554	3.00	584	3.31	*647*	*4.08*	713	4.83
7,568	2200			408	1.99	441	2.24	470	2.52	499	2.76	536	3.18	565	3.52	591	3.87	650	4.59	*711*	*5.35*
8,256	2400			433	2.42	467	2.73	492	3.04	518	3.35	551	3.73	578	4.08	603	4.42	661	5.18	715	6.01
8,944	2600					489	3.35	511	3.62	539	3.97	569	4.38	591	4.66	617	5.04	670	5.87	724	6.66
9,632	2800					511	4.05	536	4.32	562	4.67	588	5.08	610	5.42	635	5.81	683	6.56	731	7.42
10,320	3000							558	5.08	585	5.53	607	5.87	628	6.21	654	6.63	698	7.42	745	8.35
11,008	3200							588	6.04	610	6.39	632	6.73	650	7.10	676	7.60	713	8.40	757	9.33
11,696	3400							614	6.90	636	7.43	656	7.87	676	8.18	698	8.63	735	9.42	775	10.6

* Clarage Fan Co.

† **Italicized figures in each column represent maximum efficiency.**

TABLE 93

Capacities of Type-HV Centrifugal Fans with Forward-Curved Blades*

Operating at maximum efficiency. Single inlet and single width. Air density 0.075 lb per cu ft

Fan Number		Static Pressure, In. of Water ¼	⅜	½	⅝	¾	⅞	1	1¼	1½	1¾	2	2½	3
1	Cfm	604	776	863	949	1,035	1,122	1,208	1,380	1,552	1,638	1,726	1,898	2,244
	Vel	700	900	1,000	1,100	1,200	1,300	1,400	1,600	1,800	1,900	2,000	2,200	2,600
	Rpm	421	520	597	669	731	790	844	944	1,039	1,120	1,192	1,331	1,475
	Bhp	.04	.09	.13	.18	.23	.29	.36	.52	.70	.86	1.02	1.39	2.02
2	Cfm	3,440	3,440	3,784	3,784	3,784	4,128	4,472	5,160	5,504	5,848	6,192	6,880	7,568
	Vel	1,000	1,000	1,100	1,100	1,100	1,200	1,300	1,500	1,600	1,700	1,800	2,000	2,200
	Rpm	226	258	294	324	356	382	411	460	507	544	580	647	711
	Bhp	.26	.33	.50	.59	.68	.83	1.04	1.52	2.00	2.42	2.93	4.08	5.35
3	Cfm	7,740	7,740	8,514	8,514	8,514	9,288	10,062	11,610	12,384	13,158	13,932	15,480	17,028
	Vel	1,000	1,000	1,100	1,100	1,100	1,200	1,300	1,500	1,600	1,700	1,800	2,000	2,200
	Rpm	151	173	196	217	238	255	275	307	338	365	388	432	476
	Bhp	.58	.74	1.13	1.32	1.51	1.86	2.33	3.41	4.46	5.43	6.59	9.16	12.0
4	Cfm	13,770	13,770	13,770	15,147	16,524	17,901	17,901	20,655	22,032	24,786	26,163	30,294	33,048
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	111	129	146	162	179	191	204	228	250	270	289	324	353
	Bhp	1.05	1.31	1.72	2.41	3.03	3.72	4.14	5.93	7.45	9.0	12.1	17.7	23.0
5	Cfm	21,500	21,500	21,500	23,650	25,800	27,950	27,950	32,250	34,400	38,700	40,850	47,300	51,600
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	88	103	116	129	143	153	163	182	200	216	230	259	282
	Bhp	1.61	2.04	2.69	3.77	4.74	5.80	6.45	9.25	11.6	15.6	19.8	27.5	35.9
6	Cfm	30,950	30,950	30,950	34,045	37,140	40,235	40,235	46,425	49,520	55,710	58,805	68,090	74,280
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	74	86	97	108	119	128	136	152	167	180	192	216	235
	Bhp	2.32	2.94	3.87	5.42	6.82	8.36	9.30	13.2	16.7	22.5	27.1	39.6	51.7
7	Cfm	42,150	42,150	42,150	46,365	50,580	54,795	54,795	63,225	67,440	75,870	80,085	92,730	101,160
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	63	74	83	92	102	109	117	130	143	155	165	185	203
	Bhp	3.19	4.02	5.28	7.40	9.30	11.4	12.7	18.2	22.8	30.6	36.9	54.0	70.5
8	Cfm	55,000	55,000	55,000	60,500	66,000	71,500	71,500	82,500	88,000	99,000	104,500	121,000	132,000
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	55	64	73	81	89	96	102	114	125	135	144	162	177
	Bhp	4.13	5.23	6.88	9.62	12.1	14.8	16.5	23.7	29.7	39.8	48.1	70.5	92
9	Cfm	69,600	69,600	69,600	76,560	83,520	90,408	90,480	104,400	111,360	125,280	132,240	153,120	167,040
	Vel	1,000	1,000	1,000	1,100	1,200	1,300	1,300	1,500	1,600	1,800	1,900	2,200	2,400
	Rpm	49	57	65	72	79	85	91	102	111	120	128	144	157
	Bhp	5.23	6.62	8.7	12.2	15.3	18.8	20.9	30	37.6	50.5	61.0	89.0	116.5

* All values given were abstracted from tables of the form of Table 92 and represent the most efficient operation for each static pressure listed. Clarage Fan Co.

TABLE 94

General Data for Type-HV Fans*

Single-width fans. Arrangements c, FB.† All dimensions in inches

Size of Fan	Arrangements	Size of Outlet	Outside Diameter of Inlet	Extreme Dimensions for Full Housed Top Horizontal Discharge Fans			Wheel		Standard Pulley		Bearing Size	Weight, Lb
				Height	Length	Width	Diameter	Width	Diameter	Width		
1	FB	10 × 13	13	27	$21\frac{7}{8}$	$26\frac{5}{8}$	13	6	6	4	1	130
2	FB	$19\frac{3}{4} \times 25\frac{5}{8}$	27	$53\frac{3}{8}$	$41\frac{1}{8}$	$34\frac{3}{16}$	26	12	14	5	$1\frac{7}{16}$	480
3	FB	$29\frac{1}{2} \times 38\frac{3}{4}$	$40\frac{3}{8}$	$78\frac{7}{8}$	$60\frac{1}{4}$	$46\frac{1}{2}$	39	18	22	6	$1\frac{15}{16}$	1,075
4	c	$39\frac{3}{8} \times 51\frac{1}{4}$	$54\frac{1}{2}$	$93\frac{5}{16}$	$78\frac{1}{16}$	$65\frac{1}{2}$	52	24	36	7	$2\frac{7}{16}$	2,150
5	c	$49\frac{1}{4} \times 64$	68	$116\frac{5}{16}$	$97\frac{1}{8}$	78	65	30	48	8	$2\frac{15}{16}$	3,205
6	c	$59 \times 76\frac{3}{4}$	$81\frac{1}{2}$	139	$116\frac{11}{16}$	$92\frac{1}{2}$	78	36	62	10	$3\frac{11}{16}$	4,970
7	c	$68\frac{3}{4} \times 89\frac{1}{2}$	95	$162\frac{1}{4}$	$135\frac{3}{4}$	105	91	42	74	12	$3\frac{15}{16}$	6,900
8	c	$78\frac{1}{2} \times 102\frac{1}{4}$	109	$184\frac{1}{2}$	$154\frac{7}{8}$	$118\frac{1}{4}$	104	48	86	14	$4\frac{7}{16}$	10,000
9	c	$88\frac{3}{8} \times 115$	122	$209\frac{3}{16}$	$174\frac{7}{8}$	134	117	$54\frac{1}{4}$	98	18	$4\frac{15}{16}$	14,000

* Clarage Fan Co.

† Letters refer to the type of bearing and to the arrangement of bearings and drive. Arrangement FB has a single ball bearing with overhung wheel and overhung pulley for belt drive. In arrangement c the fan wheel is mounted between two babbitted bearings with overhung pulley for belt drive. Other arrangements are available in this manufacturer's line of fans.

TABLE 95

RATINGS OF DIRECT-CONNECTED TYPE-B VANEAXIAL FANS, ARRANGEMENT 4*

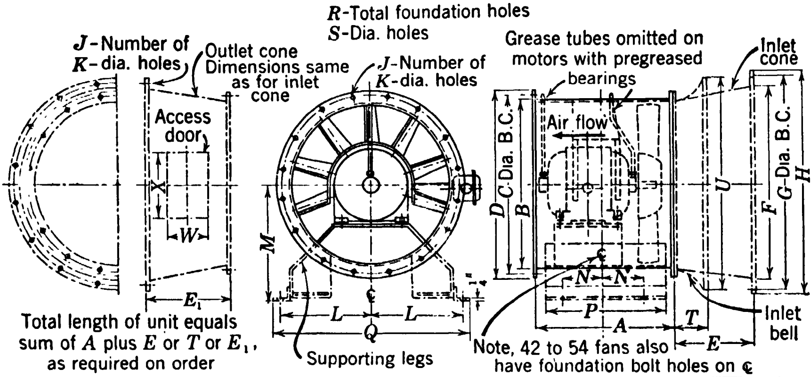
Fan Size	Speed, Rpm	Motor Size, Hp	Capacities at Various Static Pressures, Cfm								
			Free Delivery	¼-In.	½-In.	¾-In	1-In.	1-¼ In.	1½-In.	1¾-In.	2-In.
15	1750	¼	**2,820**	**2,630**	**2,220**						
18	1750	¾	**4,860**	**4,640**	**4,300**	**3,740**	**2,900**				
21	1750	1½	7,740	7,480	7,140	6,650	6,000	5,280			
	1150	½	**5,060**	**4,600**	**3,600**						
24	1750	3	11,540	11,270	10,910	10,450	9,840	9,150	8,230		
	1150	¾	**7,560**	**7,050**	**6,200**						
28	1750	7½	*18,300*	17,950	17,600	17,170	16,650	15,950	15,150	14,370	13,300
	1150	1½	12,000	11,500	10,670	9,500	7,800				
32	1150	3	17,850	17,300	16,500	15,450	14,000	12,050			
	860	1½	**13,320**	**12,450**	**11,000**	**8,250**					
36	1150	7½	25,500	24,870	24,100	23,100	21,700	20,000	18,000		
	860	3	19,050	18,200	16,700	14,500					
42	1150	15	*40,700*	*40,000*	*39,100*	*38,100*	*36,800*	*35,300*	*33,400*	*31,300*	*28,900*
	860	5	30,400	29,400	27,700	25,600	23,000	19,650			
48	860	10	*45,200*	*44,200*	*43,020*	*41,000*	*38,400*	*35,350*	*31,500*		
	690	5	36,300	34,750	32,700	29,300	24,400				
54S	860	15	57,000	54,700	52,300	50,000	47,500	45,000	42,100	39,100	35,800
	690	7½	45,500	42,800	40,000	36,700	33,200	28,900	24,000		

* Buffalo Forge Co.

Note. Horsepowers given indicate size motor suitable for 70 F., 29.92-in. Bar. conditions. Ratings in heavy type are for quiet operation, light type for moderately quiet operation, and italic type for industrial applications only. Ratings are based on tapered cone outlet. Without cone, cubic feet per minute will be reduced approximately 4 per cent with same speed and static pressure.

power are also given. Fans of this type are available in fractional sizes. Information relative to the physical dimensions of the fans listed in Table 93 is given in Table 94.

The data of Table 95 are for several different sizes of axial-flow fans. The fans for which data are included are arranged for direct connection to the driving motor and the fan speeds tabulated for each fan are the full-load speeds of commercial 60-cycle polyphase induction motors. Figure 247 includes dimensions and physical data for the fans listed in Table 95.


R-Total foundation holes
S-Dia. holes
J-Number of K-dia. holes
Outlet cone Dimensions same as for inlet cone
J-Number of K-dia. holes
Grease tubes omitted on motors with pregreased bearings
Inlet cone
Access door
W
E_1
M
L
L
Q
Supporting legs
D
C-Dia. B.C.
B
Air flow
N
N
P
A
T
E
U
F
G-Dia. B.C.
H
C
Inlet bell
Total length of unit equals sum of A plus E or T or E_1, as required on order
Note, 42 to 54 fans also have foundation bolt holes on ℄

Size	A	B	C	D	E E_1	F	G	H	J	K	L	M	N	P	Q	R	S	T	U	W	X
15	12	$15\frac{5}{32}$	$17\frac{1}{4}$	$18\frac{5}{8}$	$7\frac{1}{2}$	18	20	$21\frac{1}{4}$	12	$\frac{5}{16}$	$8\frac{1}{2}$	12	3	10	$18\frac{1}{4}$	4	$\frac{9}{16}$	3	$19\frac{1}{2}$	4	8
18	$15\frac{1}{2}$	$18\frac{3}{16}$	$20\frac{1}{4}$	$21\frac{5}{8}$	9	$21\frac{1}{2}$	$23\frac{3}{8}$	$24\frac{3}{4}$	12	$\frac{5}{16}$	10	$13\frac{1}{2}$	4	$13\frac{1}{2}$	$21\frac{1}{4}$	4	$\frac{9}{16}$	$3\frac{9}{16}$	$23\frac{1}{4}$	6	10
21	$17\frac{1}{2}$	$21\frac{7}{32}$	$23\frac{3}{8}$	$24\frac{5}{8}$	11	25	27	$28\frac{1}{4}$	16	$\frac{5}{16}$	$11\frac{3}{4}$	$15\frac{1}{2}$	5	$15\frac{1}{2}$	$25\frac{1}{4}$	4	$\frac{9}{16}$	$4\frac{3}{16}$	$27\frac{1}{4}$	6	10
24	20	$24\frac{1}{4}$	$26\frac{3}{8}$	$27\frac{5}{8}$	12	$28\frac{1}{2}$	$30\frac{3}{8}$	$31\frac{3}{4}$	16	$\frac{7}{16}$	$13\frac{1}{4}$	17	6	18	$28\frac{1}{4}$	4	$\frac{9}{16}$	$4\frac{3}{4}$	31	6	10
28	25	$28\frac{9}{32}$	$30\frac{1}{4}$	$31\frac{5}{8}$	14	$33\frac{1}{2}$	$35\frac{1}{2}$	$36\frac{3}{4}$	20	$\frac{7}{16}$	$15\frac{1}{4}$	$19\frac{1}{2}$	$7\frac{1}{4}$	$20\frac{1}{2}$	$32\frac{1}{4}$	4	$\frac{11}{16}$	$5\frac{9}{16}$	$36\frac{1}{2}$	8	12
32	$25\frac{1}{2}$	$32\frac{5}{16}$	$34\frac{3}{8}$	$35\frac{7}{8}$	16	38	40	$41\frac{3}{8}$	20	$\frac{7}{16}$	$17\frac{1}{2}$	22	$8\frac{1}{2}$	$23\frac{1}{2}$	$37\frac{1}{4}$	4	$\frac{11}{16}$	$6\frac{3}{8}$	$41\frac{1}{2}$	10	14
36	30	$36\frac{11}{32}$	$38\frac{1}{2}$	$39\frac{7}{8}$	18	43	45	$46\frac{3}{8}$	24	$\frac{7}{16}$	$19\frac{1}{2}$	25	9	$24\frac{1}{2}$	$41\frac{1}{4}$	4	$\frac{11}{16}$	$7\frac{3}{16}$	$46\frac{1}{2}$	10	14
42	35	$42\frac{3}{8}$	$44\frac{1}{2}$	$45\frac{7}{8}$	21	50	52	$53\frac{3}{8}$	32	$\frac{7}{16}$	$22\frac{3}{4}$	29	14	31	$47\frac{3}{4}$	6	$\frac{7}{8}$	$8\frac{3}{8}$	$54\frac{1}{4}$	10	14
48	39	$48\frac{7}{16}$	$50\frac{5}{8}$	$52\frac{1}{8}$	24	57	59	$60\frac{3}{8}$	32	$\frac{7}{16}$	$25\frac{3}{4}$	33	13	35	$53\frac{3}{4}$	6	$\frac{7}{8}$	$9\frac{9}{16}$	$62\frac{3}{8}$	10	14
54S	45	$54\frac{1}{2}$	57	$58\frac{1}{2}$	27	60	$62\frac{1}{2}$	64	36	$\frac{9}{16}$	$28\frac{7}{8}$	38	15	39	60	6	$\frac{7}{8}$	11	66	12	16

(*Buffalo Forge Co.*)

FIG. 247. Dimensions in inches and physical data for type-B vaneaxial fans (arrangement 4).

299. Laws of Centrifugal Fan Performance. Centrifugal fans are subject to certain physical laws. For a given fan size, duct system, and air density:

1. The capacity varies directly as the speed ratio.
2. The static pressure varies as the square of the speed ratio.
3. The speed and capacity vary as the square root of the static pressure ratio.
4. The horsepower varies as the cube of either the speed or the capacity ratio.
5. The horsepower varies as the $\frac{3}{2}$ power of the static pressure ratio.

6. The air velocity varies directly as either the speed or the capacity ratio.

As an explanation of the above statements, for the conditions specified, if the speed of a given size of fan is doubled the speed ratio becomes 2. The capacity of the fan will be doubled, the static pressure becomes 4 times as great, and the horsepower requirements to drive the fan are 8 times those before the speed was doubled.

For conditions of constant static pressure at the fan outlet the following laws hold:

7. The capacity and the horsepower vary as the square of the wheel diameter ratio.

8. The speed varies inversely as the wheel diameter ratio.

9. With a constant static pressure the speed, the capacity, and the horsepower vary inversely as the square root of the ratio of the air densities.

10. At constant capacity and speed the horsepower and static pressure vary directly as the ratio of the densities of the air.

With reference to item 9, if a fan is to deliver air against a static pressure of one inch of water with air at a density of 0.068 lb per cu ft, the speed, the capacity in cubic feet per minute and the brake horsepower are greater than when the density of the air is standard (0.075 lb per cu ft). In order to maintain a specified static pressure the fan must run faster, handle a greater quantity of air measured in cubic feet per minute, and be supplied with a greater amount of power when the air has a lessened density. Since the foregoing items all vary as the square root of the inverse density ratio, the multiplier for the table values to convert them to the actual condition of operation becomes $\sqrt{d_s \div d_a} = \sqrt{0.075 \div 0.068} = \sqrt{1.11} = 1.054$.

300. Duct Friction. The frictional resistance offered to the flow of air in ducts is dependent upon the velocity of air flow; the nature of the duct surfaces; the density of the air; the length of the duct and its bends, changes of section; and the periphery of the duct and its cross-sectional area. The cross-sectional shapes of ducts in the order of desirability from the standpoint of frictional losses are: circular, square, and rectangular. Wherever rectangular ducts are used the ratio of the width to the depth should be kept as low as possible, or the ratio of the periphery of the duct to its cross-sectional area should be minimized.

A commonly used equation to express the friction losses in a duct is

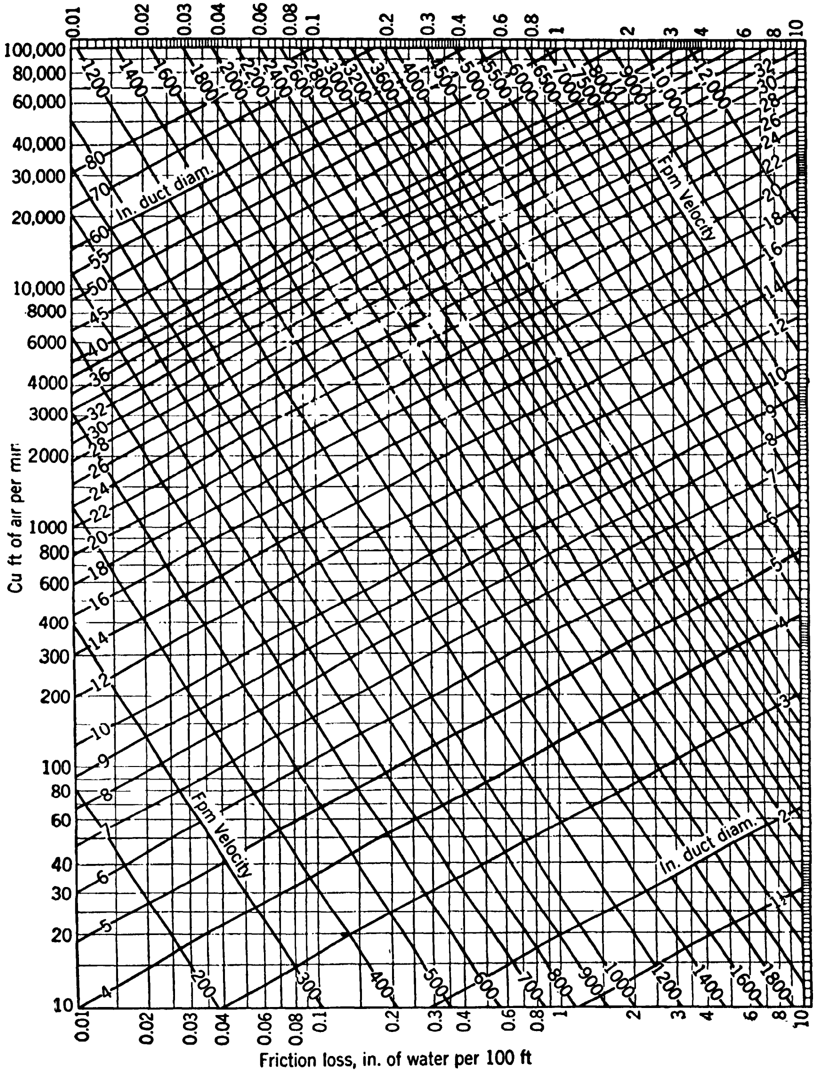
$$h_f = f \times \frac{LR}{A} \times \frac{v^2}{2g} \times \frac{12d_a}{d_w} \qquad (132)$$

where h_f = friction losses, in. of water.
f = friction factor, dimensionless.
L = length of the duct, ft.
R = perimeter of duct, ft.
A = cross-sectional area of duct, sq ft.
v = velocity of flow, fps.
d_w = density of water at the temperature of the gage fluid, lb per cu ft.
d_a = density of the air flowing, lb per cu ft.
g = acceleration due to gravity, 32.16 ft per sec per sec.

The friction factor f is dependent upon the roughness of the duct and the velocity of air flow, being greater at extremely low velocities than at higher velocities of flow where it becomes more nearly constant. An average value for commercial ducts fabricated from steel sheets and having both longitudinal and girth seams may be taken as 0.00555.

Various charts are in use for estimating the friction losses in ducts. Figure 248 illustrates a chart which gives the friction losses per 100 ft for clean round galvanized metal ducts having approximately forty joints per 100 ft. Friction loss per 100 ft. of duct as read from Fig. 248 will not always check with that calculated for the same length of the same duct by the use of formula 132, as the chart is not based on a constant value for f. The chart is for air at 69.41 F and 29.92 in. of mercury absolute pressure having a density of 0.075 lb per cu ft. When air of greater or less density is handled the chart is used as though the volume were at 69.41 F and 29.92 in. of mercury, and the friction loss as given by the chart is corrected by multiplying it by the ratio of the actual density of the air handled to 0.075. For the average application, values obtained from the chart are sufficiently accurate without correction for any air temperature between 50 F and 90 F, for any relative humidity, and, except for cities at a high elevation above sea level, for any normal variation in barometric pressure. The chart should not be used to obtain values below those shown, by extrapolation, because critical flow would occur in this region, and the data would be unreliable. For unusually rough pipes, information beyond the scope of this book should be sought, as the friction loss may be more than double that obtained by use of the chart.

The friction losses of a rectangular duct may be calculated by means of equation 132, provided that a suitable value of f is available. When the data of Fig. 248 are to be used in connection with a rectangular duct it is first necessary to determine the equivalent diameter, in



(*From "Heating Ventilating and Air Conditioning Guide 1948." Used by permission.*)

FIG. 248. Friction-pressure losses of air in circular ducts. Based on standard air at 29.921 in. of mercury barometric pressure and 69.41 F flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.

TABLE 96

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION*

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5

28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.0	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

* Additional sizes: 4 × 5 = 4.9, 4 × 6 = 5.4, 4 × 7 = 5.8, 5 × 5 = 5.5, 5 × 6 = 6.3, and 5 × 7 = 6.5.

inches, of a round duct having the same friction losses per foot of length. Such data are given in Table 96. When the circular-duct-diameter equivalent d of a rectangular duct whose dimensions a and b are not included in Table 96 is to be determined, use may be made of equation 133.

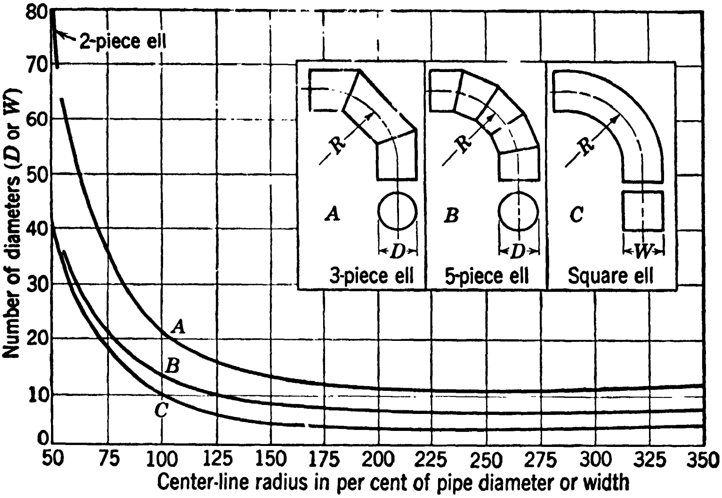
$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a + b}} \tag{133}$$

Equation 133 is derived from equation 132, and d, the equivalent diameter, is in either feet or inches, depending upon the dimensional units used. The assumption is made that h_f, f, L, and Q, the air flowing in cubic feet per second, are the same in both the circular and the rectangular duct under consideration. The friction losses in a duct represent static pressure which the fan must create and which is used up as the air flows.

301. Effect of Entrance, Elbows, and Changes of Duct Section on Friction and Turbulence Losses. Air entering a duct from an enlarged distributing chamber, plenum chamber, or from the outside air occasions a loss of pressure which is known as an entrance loss. The entrance loss is greatly affected by the shape of the entrance and may be anywhere from 0.1 to 0.9 of the velocity pressure h_v of the air flowing in the duct. A bell-mouthed entrance offers the minimum resistance, whereas the entrance loss is the greatest when the pipe extends through the wall of the plenum chamber. When louvers are used the total entrance loss will usually be between $1.25h_v$ and $2.0h_v$, depending upon the design. Here h_v is based on the free area velocity through the louvers. Common practice is to use $1.5h_v$ as the entrance loss where the inlet to the system is equipped with louvers. Pressure losses in elbows are also important and should be taken into account in the design. The pressure losses due to elbows are dependent upon the shapes of the duct sections, their radii of curvature, and the velocities of flow. The effect of elbows may be allowed for by adding to the straight length of pipe an equivalent length as an allowance for each elbow. These allowances expressed as some multiplier times the duct diameter or width, in feet, are dependent upon the ratio of the radius of curvature to the duct diameter or width.

Figure 249 gives three curves which may be used as an indication of the number of duct diameters or widths that should be added to the length of straight duct to allow for the friction loss in each elbow in the system. The curves may be used for either round or rectangular ducts. Curve *A* should be used if the quality of construction is questionable, even if the elbow is type *B* or type *C*. It may be noted from the curves in Fig. 249 that little advantage is to be gained by using a centerline radius that is greater than 2 times the duct diameter or width, but whenever possible the ratio of centerline radius to duct diameter or width should be at least 1 and preferably 1.5.

Where space considerations make it necessary to use short radii or miter elbows in rectangular or square-duct work, pressure losses may be greatly reduced by the use of turning vanes as shown in Table 97.



(*From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.*)

FIG. 249. Straight-pipe equivalents of elbows for equal friction losses.

Another method of allowance is to express the loss of pressure in an elbow or other obstruction in terms of the velocity head existent in the elbow. Table 98 gives data for the multipliers of the actual velocity pressures of elbows which are used to express the pressure losses in inches of water.

Turbulence losses will occur wherever the cross-sectional area of the duct is changed[1] and may amount to as much as 60 per cent of the change in velocity pressure unless the two different sized parts of the system are connected by a gradually tapered transition section. Therefore, abrupt changes in area or in shape should be avoided in air duct design wherever possible, and where such changes cannot be

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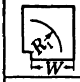
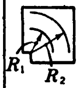
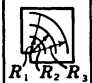
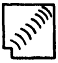
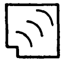
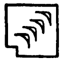
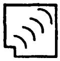
[1] "Pressure Losses Due to Changes in the Cross-Sectional Area in Air Ducts," by A. P. Kratz and J. R. Fellows, *University of Illinois Engineering Experiment Station Bulletin* 300.

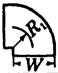
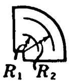
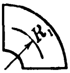
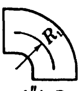
TABLE 97*

EFFECT OF VANES ON PRESSURE LOSS OF 7-IN. SQ. VENTILATING DUCT†

Expressed in feet of total equivalent length of duct (ELD)

Square-miter elbow								
	Radius Ratio	$\frac{R_1}{W}$	0.0	0.2	0.4	0.6	0.8	1.0
	ELD, ft		41.1	30.5	27.5	30.1	37.7	38.5
	Radius Ratio	R_1/W	0.0	0.2	0.3			
		R_2/W	0.0	0.4	0.5			
	ELD, ft		41.1	23.5	23.3			
	Radius Ratio	R_1/W	0.0	0.2	0.3			
		R_2/W	0.0	0.4	0.5			
		R_3/W	0.0	0.6	0.7			
	ELD, ft		41.1	20.7	22.2			

				
Vane	*A*	*B*	*C*	*D*
ELD, ft	21.8	17.0	17.9	17.8

Standard elbows with various radii								
	Radius Ratio	$\frac{R_1}{W}$	0.0	0.2	0.4	0.6	0.8	1 0
	ELD, ft		39.7	23.3	22 0	25.7	28.9	39.7
	Radius Ratio	R_1/W	0.0	0.2	0.3	0.4	0.5	0.6
		R_2/W	0.0	0.4	0.5	0.6	0.7	0.8
	ELD, ft		39.7	20.0	22.0	23.0	23.8	25.7
 1½" I. R.	Radius Ratio	$\frac{R_1}{W}$	0.0	0.4	0.6	0.8	1.0	1.2
	ELD, ft		25.3	17.7	16.5	18.7	23.5	25.6
 4" I. R.	Radius Ratio	$\frac{R_1}{W}$	0.0	0.7	0.8	0.9	1.0	1.2
	ELD, ft		14.2	13.3	13.0	12.7	12.5	12.7

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* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

† For more complete data see ASHVE Research Report 1216, "Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct," by M. C. Stuart, C. F. Warner and W. C. Roberts (*ASHVE Trans*, 1942. Vol. 48.)

Note A: Vane *A* made up of a large number of small splitters; *B* made up of a small number of large splitters bent on a large radius; *C* hollow vanes having different outside and inside curvature; and *D* four splitters with $R/W = 0.4$. Elbow same as *D* except 2 in. trailing edge on the end of each splitter, ELD in feet = 17.0.

Note B: The air velocity has no effect on the loss of elbows when the loss is expressed as equivalent length of duct.

TABLE 98

MULTIPLIERS OF VELOCITY PRESSURE FOR ESTIMATING THE PRESSURE LOSSES IN ELBOWS OF CIRCULAR AND RECTANGULAR CROSS SECTION

Ratio of Centerline Radius to Diameter or Width of Duct	Elbows of Circular Cross Section, Diameter = *D*	Elbows of Rectangular Cross Section, Width = *W*	Ratio of Centerline Radius to Diameter or Width of Duct	Elbows of Circular Cross Section, Diameter = *D*	Elbows of Rectangular Cross Section, Width = *W*
0	0.87	1.2	1.25	0.20	0.12
0.25	0.81	1.1	1.50	0.17	0.09
0.50	0.76	0.95	1.75	0.16	0.08
0.75	0.38	0.33	2.00	0.15	0.08
1.00	0.26	0.18	2.25	0.14	0.07

avoided proper allowance for the turbulence loss occurring in them should be made.

The following allowances of Table 99 may be used, when branches are taken from a main duct, as entry losses into the branches and also for other resistances to flow.

TABLE 99

MULTIPLIERS OF VELOCITY PRESSURE FOR ESTIMATING ENTRY LOSSES AND RESISTANCES TO AIR FLOW

Obstruction		Multiplier		Obstruction	Multiplier
Entrance to branch taken at an angle from the center line of the main duct	15°	0.10	$\times h_v$ in branch	Abrupt entrance	0.45 to 0.87
	30°	0.18		Coned entrance	0.23
	45°	0.23		Registers. Free	
	60°	0.45		area = ½ gross	1.25 to 1.5
Square tee at end of duct		1.0		area = area of duct	h_v in free area

302. Types of Duct Systems. There are two schemes for the placement of air distributing ducts, namely, the individual and the trunk-

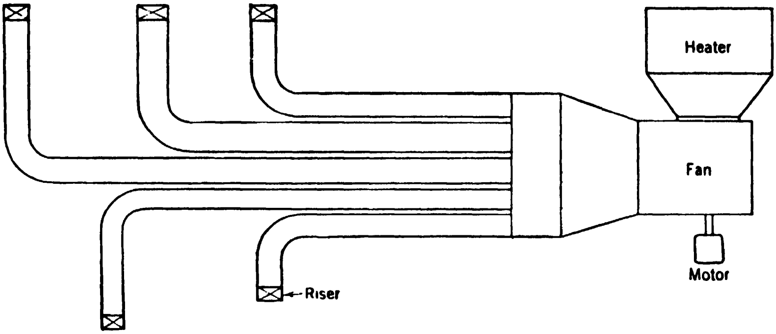


FIG. 250. Individual duct system.

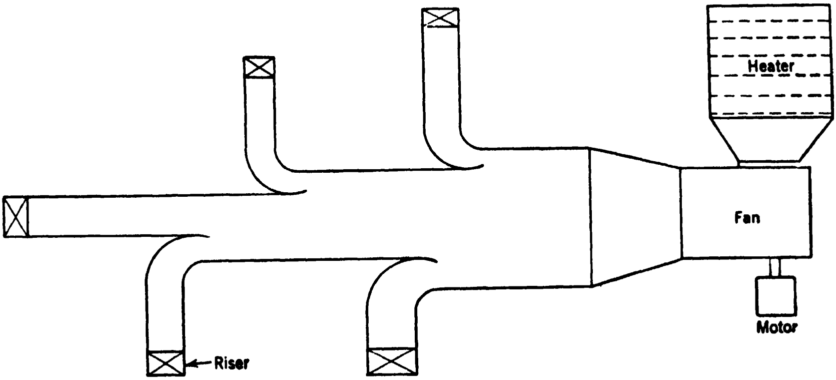


FIG. 251. Trunk duct system.

line systems. The individual system, Fig. 250, provides a separate duct to each room from a common plenum chamber into which the fan discharges. The trunk-line system, Fig. 251, has one or more main ducts from which branches are taken to serve the various spaces.

The design of either system may be based on the quantities of air to be handled and the allowable velocities with which the air may flow in the various sections of the system. Recommended velocities of air flow in different portions of the system are as given by Table 100.

TABLE 100

AIR VELOCITIES FOR DUCTS AND RISERS

	Feet per Minute	
Service	Public Buildings	Industrial Plants
Air intakes from the outside	600 to 900	1000 to 1200
Air washers	500	500 to 600
Heater connections to fan	700 to 900	1000 to 1400
Main ducts	1000 to 1500	1200 to 2400
Branch ducts and risers	500 to 600	900 to 1800
Supply registers and grilles	250 to 450	
Supply openings		350 to 500
Supply grilles near the floor	150 to 250	

This scheme of design makes the calculation of the pressure losses in the separate parts of the system laborious as the friction losses are not constant when the ducts are proportioned on the basis of allowable velocities of flow. The easier method of sizing the ducts of a system is to determine the size of the last outlet of a duct on the basis of the allowable outlet velocity and the quantity of air to be handled. From the size so determined and the quantity of air discharged the friction losses per 100 ft of duct may be ascertained and the system designed using this value as the constant friction pressure loss. When the design is so made, the velocities of air flow are not uniform but are greatest near the fan outlet.

With either the individual or the trunk arrangement of ducts the fan must create enough pressure to discharge the air with the required quantity and velocity at the outlet farthest removed from the fan or the outlet which has the greatest resistance to flow between it and the fan. This usually but not always means the longest run of duct. The other ducts are either so proportioned that the available head is used up within them or they are dampered to give additional resistance so that the proper quantity of air will pass through them.

303. Trunk-Duct Design Procedure and Fan Selection. The system should be laid out to deliver the air with the least expenditure of power, materials, and space, keeping the design simple and avoiding obstructions. Sharp elbows and bends are to be avoided, and the arrangement of the duct outlets should be such as will insure the proper distribution of air. The size of each outlet should be based on the quantity of air to be delivered and a suitable outlet velocity. Either the velocity method or the friction-pressure-loss method can be used in proportioning the duct sizes. The latter method is the simpler and more satisfactory. After the sizing of the ducts is completed, the static pressure against which the fan must operate should be calculated. The following will serve to illustrate the layout of a duct system by the friction-pressure-loss method.

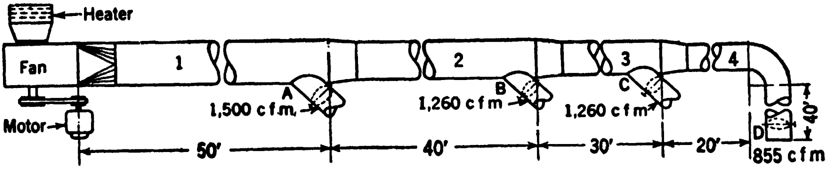


FIG. 252. Fan heating system.

Example. A fan discharges air through a system of circular ducts, as shown by Fig. 252, at an average temperature of 130 F at the outlets when the barometric pressure is 29.92 in. of mercury. The building heat losses are 330,800 Btu per hr when the air of the space is maintained at 60 F. The quantities of air delivered at the outlets are as indicated by Fig. 252. The outlet air velocities are approximately 800 fpm. All outlets are to be dampered to control the amount of air flowing. The ratio of the elbow radius to the duct diameter is 1.5. The actual friction losses at the heater and the fan inlet are 0.388 in. of water. Find the sizes of the duct and select a fan.

Solution. The weight of air handled per hour is 330,800 ÷ 0.24 (130-60) = 19,690 lb, and the volume of the air as delivered is 19,690 ÷ (60 × 0.06729) = 4875 cfm at 130 F. The sizes of the duct outlets are:

$$A,\ (1500 \times 144) \div 800 = 270 \text{ sq in. or } 18.5 \text{ in. diameter}$$

$$B \text{ and } C,\ (1260 \times 144) \div 800 = 226.8 \text{ sq in. or } 17 \text{ in. diameter}$$

$$D,\ (855 \times 144) \div 800 = 154 \text{ sq in. or } 14 \text{ in. diameter}$$

The friction-pressure-loss chart, Fig. 248, shows that a duct 14 in. in diameter handling 855 cfm at 69.41 F will have friction-pressure losses amounting to 0.067 in. of water per 100 ft of run. Considering this value a constant friction loss, the sizes of the other portions of the main duct are determined to be as indicated in the following table:

Section	Cubic Feet per Minute	Duct Diameter, In.	Velocity of Air Flow, Ft per Min	Length of Duct Section, Ft
4	855	14	800	60
3	2115	20	970	30
2	3375	23.5	1120	40
1	4875	27	1225	50
				Total 180 ft

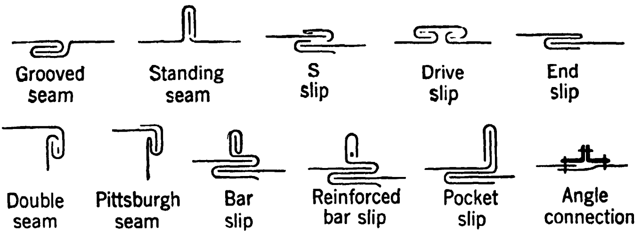
The actual friction pressure losses in the duct length of 180 ft are

$$\frac{0.067 \times 180}{100} \times \frac{0.06729}{0.075} = 0.108 \text{ in. of water}$$

The velocity pressure in the elbow of section 4 and at the outlet of section 4 is $(13.2^2 \times 12 \times 0.06729) \div (2 \times 32.16 \times 62.3) = 0.036$ in. of water. The pressure loss in the elbow of section 4 equals $0.036 \times 0.17 = 0.006$ in. of water. The total pressure loss in the discharge duct $= 0.108 + 0.006 = 0.114$ in. of water. The total resistance of the system $= 0.114 + 0.388 = 0.502$ in. of water. This is also the static pressure against which the fan must operate.

The volume of air handled under actual conditions is 4875 cfm at 130 F and 29.92 in. of mercury. Under rating conditions the fan would have to handle $4875 \div \sqrt{0.075 \div 0.06729} = 4875 \div 1.06 = 4600$ cfm (fan law 9). Table 93 does not include a combination of a static pressure of 0.5 in. and a volume of 4600 cfm. However, the table indicates that fan 2 will deliver 3784 cfm of standard air against $\frac{1}{2}$ in. of static pressure when operated at a speed of 294 rpm. Under operating conditions with air density = 0.06729 lb per cu ft the fan would deliver $3784 \times 1.06 = 4020$ cfm against 0.5 in. static pressure and would require a speed of $294 \times 1.06 = 312$ rpm and $0.50 \times 1.06 = 0.53$ bhp to drive it. Since the volume required is 4875 cfm the speed will have to be increased. The required speed is $312 \times 4875 \div 4020 = 378$ rpm (fan law 1), the available static pressure is $0.5 \times (\frac{378}{312})^2 = 0.736$ in. of water (fan law 2), and the required horsepower is $0.53 \times (\frac{378}{312})^3 = 0.95$ (fan law 3). It is possible that examination of a more complete table showing data for the fractional sizes which have been mentioned would disclose a fan whose static-pressure characteristic is better suited to this particular application. It is also possible that examination of tabular data for a line of fans made by some other manufacturer might disclose a fan which would more nearly provide the exact combination of volume and static pressure that is needed in this installation. It may be noted from a study of Table 95 that axial-flow fan size 18, when operated at 1750 rpm, will deliver 4300 cfm of standard air against $\frac{1}{2}$-in. static pressure and requires approximately 0.75 bhp. Under operating conditions this fan would deliver $4300 \times 1.06 = 4560$ cfm when operated at $1750 \times 1.06 = 1855$ rpm and when supplied with $0.75 \times 1.06 = 0.795$ bhp. The actual speed required would be $1855 \times \frac{4875}{4560} = 1980$ rpm, and the required horsepower would be $0.795 \times (\frac{1980}{1855})^3 = 0.966$. This fan is therefore capable of meeting the requirements of this system, but it would require slightly more power than would be required by the number 2 fan selected from Table 93. A practical difficulty in using one of the axial-flow fans for which data are listed in Table 95 is that they are direct connected to constant-speed motors so that the speed could not be changed to meet the exact delivery that was specified. Therefore, if the exact requirements as to volume delivery are to be met, the choice from all the tables available in this text is the number 2 fan from Table 93.

In the system of the foregoing example, part of the velocity pressure of the air at the fan outlet would be converted into static pressure because of the reduction in velocity as the main stream flows from the fan to the discharge end of section 4. The theoretical maximum static pressure regain in such a system is the difference between the velocity pressures at the fan discharge and that in section 4. The theoretical maximum regain never occurs in practice because of turbulence losses as the velocity of the air is decreased. Some designers recommend that the required static pressure be taken as the estimated friction loss in the entire system minus one-half the theoretical maximum static-pressure regain. However, it is the usual practice to neglect conversion of velocity pressure to useful static pressure and select fans on the basis of required fan static pressure being equal to the sum of all of the estimated friction losses.



(*From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.*)

FIG. 253. Forms of seams and joints in sheet-metal duct construction.

304. Air-Duct Construction. All duct construction should be as smooth as possible where the air passes over its inner surfaces. The ducts should be airtight and rigid to prevent vibration of the sheets when constructed of galvanized sheet steel. Gages of sheet steel are given in Table 101.

Figure 253 shows several methods of joining two edges in duct construction. Straight sections of round duct are usually formed by rolling the sheets to the proper radius and joining the two edges with a grooved seam. Rectangular-duct sections may be made by breaking the corners of flat sheets cut to the proper width then joining the

TABLE 101[*]

RECOMMENDED SHEET-METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[†]

U. S. Standard Gage	Maximum Side, In.	Type of Transverse Joint Connections[‡]	Bracing
26	Up to 12	S, drive, pocket or bar slips, on 7 ft. 10 in. centers	None
24	13 to 24	S, drive, pocket or bar slips, on 7 ft 10 in. centers	None
	25 to 30	S, drive, 1-in. pocket or 1-in. bar slips, on 7 ft 10 in. centers[§]	$1 \times 1 \times \frac{1}{8}$ in. angles 4 ft from joint
22	31 to 40	drive, 1-in. pocket or 1-in. bar slips, on 7 ft 10 in. centers[§]	$1 \times 1 \times \frac{1}{8}$ in. angles 4 ft from joint
	41 to 60	$1\frac{1}{2}$-in. angle connections, or $1\frac{1}{2}$-in. pocket or $1\frac{1}{2}$-in. bar slips with $1\frac{3}{8} \times \frac{1}{8}$ in. bar reinforcing on 7 ft 10 in. centers[§]	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in. angles 4 ft from joint
20	61 to 90	$1\frac{1}{2}$-in. angle connections, or $1\frac{1}{2}$-in. pocket or $1\frac{1}{2}$-in. bar slips 3 ft 9 in. maximum centers with $1\frac{3}{8} \times \frac{1}{8}$ in. bar reinforcing	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in. diagonal angles or, $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in. angles 2 ft from joint
18	91 and up	2-in. angle connections or $1\frac{1}{2}$-in. pocket or $1\frac{1}{2}$-in. bar slips 3 ft 9 in. maximum centers with $1\frac{3}{8} \times \frac{1}{8}$ in. bar reinforcing[‖]	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in. diagonal angles, or $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in. angles 2 ft from joint

[*] From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

[†] For normal pressures and velocities utilized in typical ventilating and air-conditioning systems. Where special rigidity or stiffness is required, ducts should be constructed of metal two gages heavier. All uninsulated ducts 18 in. and larger should be cross-broken. Cross-breaking may be omitted on uninsulated ducts if two gages of heavier metal are used.

[‡] Other joint connections of equivalent mechanical strength and air tightness may be used.

[§] Duct sections of 3 ft 9 in. may be used with bracing angles omitted, instead of 7 ft 10 in. lengths with joints indicated.

[‖] Ducts 91 in. and larger require special field study for hanging and supporting methods.

edges by means of either grooved or standing seams. Sections of elbows or transformation pieces are usually formed with Pittsburgh corner seams. Complicated fittings are usually constructed with double-seam corners. Table 101 indicates the applications of the various slips and connections which may be used for securing transverse joints in rectangular ducts and suggests bracing for the sides of rectangular duct sections.

305. Duct Insulation. All ducts in heating, ventilating, and air-conditioning systems should be insulated against transfer of heat to or from the conditioned air when the ambient temperature surrounding them is appreciably more or less than that of the air flowing. When no insulation is applied to sheet-metal ducts the principal resistances to heat transmission are the inside- and the outside-air films, as the resistance of the metal is negligible. An average value for the conductance of the outside film = 1.5 Btu per sq ft per hr per deg F temp difference, and this value can be applied to practically all installations. The conductance of the inside film depends principally on the velocity of the air and the diameter of the duct. The following equation may be used:

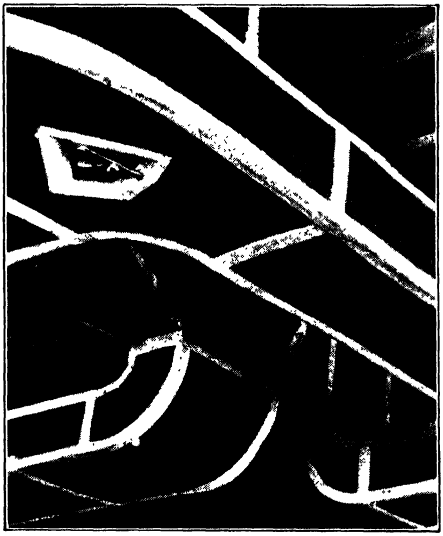
$$f_i = 0.32 v_a^{0.8} \div D^{0.25} \qquad (134)$$

where f_i = film conductance, Btu per hr per sq ft per deg F temp difference.

v_a = average velocity, fps.

D = inside diameter of duct, ft.

The overall coefficient U and the heat transfer in Btu per hour, either with or without insulation, may be estimated by the method that is outlined in Chap. 4 for the calculation of heat losses through building walls.



(Grant-Wilson Co., Inc.)

FIG. 254. Exterior insulation applied to sheet-metal ducts.

Any insulating material in suitable form for application to ducts may be used and for large ducts may be applied to either the inside or the outside surface. However, any insulating material applied to the inside surface of ducts should be non-combustible because of the possibility of "duct fires." Non-combustible insulating material applied to the inside of ducts is helpful in reducing noise, but great care must be used in its application to secure the smoothest possible surface, and the character of the final interior surface must be taken into consideration in estimating the friction loss of the system. Insulating material in blanket form may be cut into pieces of desired shapes and placed as shown in Fig. 254. In applying this type of insulation the ducts are usually first covered with glue which firmly cements the insulating material to the metal. Corners of the insulated duct and joints in the covering are usually covered with special tape as shown in the illustration.

306. Measurement and Specification of Noise Levels. Specifications in regard to noise produced by machinery or by air flow and transmitted to occupied spaces are now being included in many contracts covering the installation of air-conditioning equipment. Noise specifications are usually expressed in terms of decibels. The decibel is a unit used to express the relation between two amounts of power. The difference in decibels between a power P_1 and a larger power $P_2 = 10 \log_{10} (P_2 \div P_1)$. In sound measurement the threshold of hearing has been established as 10^{-16} watts per square centimeter. The intensity level of any sound is then given by the following equation.

$$I = 10 \log_{10} \frac{P}{10^{-16}} \qquad (135)$$

where I = sound intensity level, db.
P = power of the sound waves, watts per sq cm.

One decibel is about the smallest change in sound level that can be perceived by the human ear.

It is most satisfactory to measure noise with a sound-level meter called an acoustimeter which is now available for this purpose. An acoustimeter usually consists of a microphone, a high-gain audio-amplifier, and a rectifying milliammeter. The instrument is usually designed to indicate sound intensities directly in decibels.

Since a sound intensity is 10 times the logarithm of a ratio of two powers and not a definite power, the intensity of a combination of two different sounds superimposed one on the other cannot be predicted by adding the two separate sound levels. In general it has been found that combining two separate sounds of equal powers results in a combined intensity that is 3 db higher than that of the individual noises regardless of the actual levels.

It is necessary in specifying maximum noise levels from machines to agree on the exact location of the microphone of the acoustimeter when the acceptance test is made. Table 102 gives noise levels in decibels for several situations which have been experienced by most people. Exact specifications of permissible noise levels in air-conditioning work are as yet not in common use. In general, air-condition-

TABLE 102*

Typical Noise Levels†

Rooms	Noise Level in Decibels to Be Anticipated		
	Minimum	Representative	Maximum
Sound-film studios	10	14	20
Radio-broadcasting studios	10	14	20
Planetarium	15	20	25
Residence, apartments, etc.	33	40	48
Theaters, legitimate	25	30	35
Theaters, motion-picture	30	35	40
Auditoriums, concert halls, etc.	25	30	40
Churches	25	30	35
Executive offices, acoustically treated private offices	30	38	45
Private offices, acoustically untreated	35	43	50
General offices	50	60	70
Hospitals	25	40	55
Classrooms	30	35	45
Libraries, museums, and art galleries	30	40	45
Public buildings, post offices, etc.	45	55	60
Courtrooms	30	35	45
Small stores	40	50	60
Upper floors department stores	40	50	55
Stores, general, including main floor department stores	50	60	70
Hotel dining rooms	40	50	60
Restaurants and cafeterias	50	60	70
Banking rooms	50	55	60
Factories	65	77	90
Office machine rooms	60	70	80
Vehicles			
Railroad coach	60‡	70	80
Pullman car	55‡	65	75
Automobile	50	65	80
Vehicular tunnel	75	85	95
Airplane	75	80	90

* From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

† These values are tentative. More detailed measurements by D. F. Seacord, Bell Telephone Laboratories (*Journal Acoustical Society of America*, Vol 12, pp. 183–187, 1940) give average values and standard deviations of room noise in residences, offices, stores, factories, etc., in large American cities.

‡ For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

ing apparatus is regarded as acceptable from a noise standpoint if its operation does not increase the original noise level of the conditioned spaces by more than 3 db in places that are already moderately noisy. This means that the equipment noise alone must not be greater than the room noise alone.

307. Control of Noise Level. A thorough discussion of the various treatments which may be used to reduce noise is beyond the scope of this book. Transmission of mechanical noises from either the fan or the motor through the duct system can be prevented by use of either a canvas or an asbestos cloth to connect the fan inlet and the fan outlet to the respective parts of the system. Transmission of vibration noises through the building itself can be prevented by mounting the apparatus on pads of resilient material such as either rubber or cork. In general, rotational or vortex noises from a fan will be at acceptable levels if a fan is selected such that it will operate at or near its maximum mechanical efficiency. Noises due to the flow of air through the system can be kept down to acceptable levels without the use of sound insulation by proportioning all ducts so that the velocity will not exceed 900 to 1200 fpm in main ducts or 600 to 1000 fpm in branches. Air velocity leaving registers or grilles should not exceed 200 to 400 fpm where quiet operation is required.

When practical consideration demands the use of a system which will produce a noise level that is above that permissible in the space to be conditioned, information on the use of sound absorbers should be sought.

308. The Problem of Air Distribution. Regardless of how well the rest of the equipment may be designed the performance of heating, ventilating, or air-conditioning systems will be unsatisfactory unless suitable provisions are made for the uniform distribution of the heated air, the fresh air, or the conditioned air throughout the space that is being treated. The problem of securing proper distribution in large rooms without producing objectionable drafts in local areas is sometimes a difficult one. Air movements must be limited to 30 fpm in all spaces that may be occupied by people who are seated if ideal conditions are to be maintained. A somewhat higher velocity may not be noticed if the temperature and relative humidity of the air both fall within the upper limits of the comfort zone. The average person, as usually clothed for a public gathering, is especially sensitive to air movements over the neck or the ankles. In air-conditioned spaces which are to be used for purposes such that the occupants will always be moving about, velocities up to 120 fpm may be tolerated without the feeling of discomfort.

Perfect distribution would be achieved by a system in which the air is introduced at room temperature through one entire room surface, uniformly perforated, and removed through the opposite room surface perforated in a similar manner. However, practical considerations require the use of air diffusers such as registers or grilles which occupy only a small portion of the room surface in which they are placed. Because of the limited area of the distribution outlets the outlet velocity required for the delivery of the required amount of air is usually several times the velocity which can be tolerated by occupants without discomfort. Even if it were practicable to introduce the conditioned air at velocities that would not be noticed by occupants close to the points of entry, distribution of the air throughout the room might be unsatisfactory due to natural circulation caused by differences in temperature between the conditioned air and that already within the enclosure. Fortunately, the movement of air withdrawn from a room through grille openings is not noticeable to occupants seated near them even when the free-area velocity is as high as 750 fpm. It is therefore possible to project the conditioned air into the conditioned space at velocities which are high enough to secure satisfactory distribution through specially designed diffusers located in the ceiling or in the upper part of the side walls and then remove it through grilles which are placed in the floor or in the side walls close to floor level. Definite schemes are described in detail in Arts. 330 to 341 inclusive.

309. Behavior of a High-Velocity Jet of Air. A stream of air introduced into a room with considerable velocity of flow entrains room air adjacent to it. This action causes the static pressure of the air near the jet to be lessened, and, as a result, room air moves to produce a secondary circulation as indicated by the arrows of Fig. 255. This secondary circulation is sometimes advantageous when the temperature of the jet of air is above or below the room temperature, as it produces a mixing process that is desirable. The same "dragging effect" which causes the room air adjacent to the jet to be carried along with it also causes the outer edge of the jet to be pulled off to the side, thus causing a gradual "decay" of the projected stream. The "throw" of an air jet may be defined as the distance between

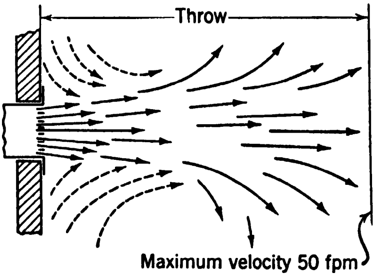
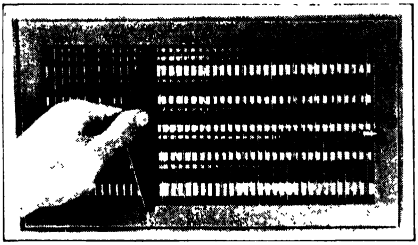


FIG. 255. Secondary air circulation produced by jet action.

the outlet and a position at which air motion has been reduced to a maximum velocity of 50 fpm. The "throw" depends upon the initial velocity and upon the initial size and shape of the diffuser. In large rooms with high ceilings such as auditoriums the diffusers may be few and large, while in low-ceilinged rooms such as railroad cars a perforated type of diffuser may be required in order that the projected streams may "decay" completely before reaching the heads of occupants only a few feet away.

310. Diffusion Devices. Several different arrangements have been developed for the purpose of providing in one device an architecturally acceptable mask for the end of the supply duct and a control of the stream of conditioned air which issues from it. A few of the more commonly used types will be discussed in the following paragraphs.



(*United States Register Co.*)

FIG. 256. Outlet grille fitted with horizontal and vertical air-directional blades.

Figure 256 shows a grille with two sets of directional vanes. The vertical vanes may be turned by use of a special tool to adjust the spread of the jet in a horizontal direction and a second set back of the first which is used to regulate the angle of the jet in a vertical plane. The horizontal vanes at the rear are operated by the handle at the right end of the grille. Grilles of this same general type are also made with only one set of adjustable vanes while still another design includes only fixed vanes designed to achieve a predetermined directional effect.

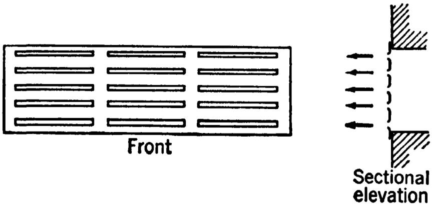


FIG. 257. Injection-type air-flow grille.

A special type of grille shown in Fig. 257 is designed to create maximum secondary circulation and is very useful where it is necessary to introduce air at temperatures considerably below the comfort level. With this type of grille properly placed, incoming air may have a temperature as low as 25 deg F below room temperature without danger of creating cold drafts, provided that the grilles are placed at a sufficient distance from the occupants of the room.

Figure 258 is an illustration of the use of a diffuser which consists of several specially formed concentric rings. The diffuser shown is designed for installation in ceilings and is arranged in such a way that all the air discharged is directed downward at some definite angle with the horizontal. The purpose of this arrangement is to uniformly deliver the conditioned air to the floor area which is served by each outlet. This type of diffuser may be obtained in a variety of vane arrangements so as to provide "blows" in one, two, three, or four directions. Suitable arrangements may be placed in a side wall instead of in the ceiling.

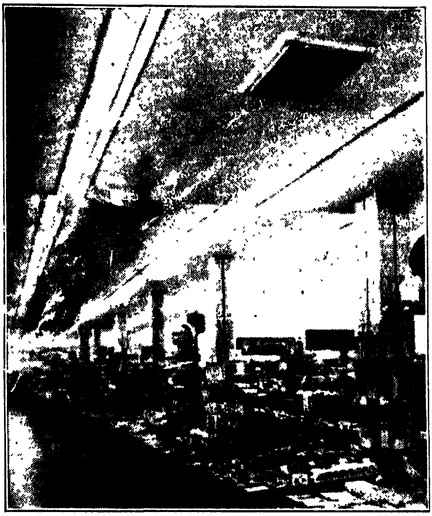
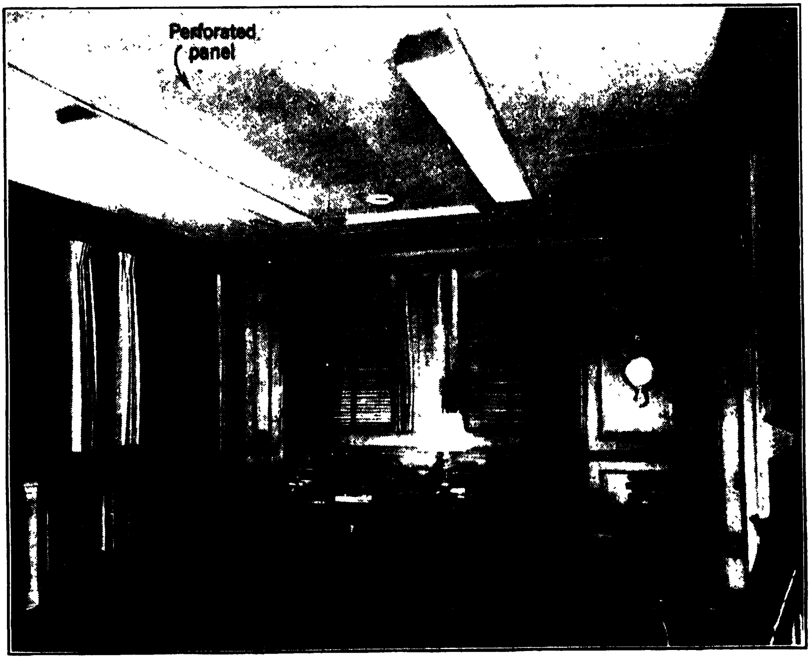

(*Air Devices, Inc.*)

FIG. 258. Concentric-ring air diffuser.

Figure 259 shows an example of a perforated type of diffuser



(*Pyle National Co.*)

FIG. 259. Perforated-panel air diffuser.

which is well adapted to use in rooms where satisfactory results could not be achieved with any of the diffusers which have been previously discussed due to low ceilings. Continuous slots arranged to discharge the conditioned air in a horizontal direction near the ceiling have also been used with good results in low-ceilinged rooms which are not more than 40 ft in width. An advantage in using the perforated type of diffuser shown in Fig. 259 is that it is not readily discernible when incorporated in acoustical ceilings.

In regard to any outlet register or grille, the ratio of the length to the width of its core is called its aspect ratio.

311. Automatic Static-Pressure Control in Duct Systems. The velocity of the air passing through the free area of a diffuser will be proportional to the square root of the total pressure in the duct immediately back of the diffuser if the diffuser is placed at the end of the duct and proportional to the square root of the static pressure only if the diffuser is placed in a side of the duct. In any case the velocity pressure is usually small when compared with the static pressure; so, in general, it may be said that the velocity of air through a diffuser is determined by the static pressure in the duct just back of it. In a system where several rooms are served by a single duct, occupants of certain rooms may alter damper settings in such a way that the static pressure in other portions of the duct may be changed causing the delivery of air to those rooms to be increased or decreased to the point where discomfort of the occupants is a result. Unsatisfactory delivery of conditioned air to specific rooms of a system due to the cause that has been discussed can be prevented by the installation of an automatic static-pressure controller working on the principle illustrated in Fig. 260.

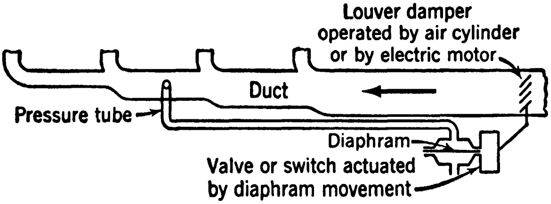

Louver damper operated by air cylinder or by electric motor
Duct
Pressure tube
Diaphram
Valve or switch actuated by diaphram movement

FIG. 260. Flow control by static-pressure tube in duct system.

312. Air Removal. Provision must be made for the escape of a volume of air approximately equal to that of the supply air to avoid building up a noticeable pressure in a conditioned space which would interfere with the closing of doors. In some types of public buildings where exterior doors are in almost constant use due to continuous in and out traffic it may not be necessary to install either grilles or ducts for exhausting air from the room. However, in most air-conditioning systems it is necessary to provide outlet grilles and ducts for conducting the exhausted air either to the outside of the building or to the air-conditioning apparatus for recirculation to the space served.

Outlet grilles are usually simple grids intended merely to mask the opening and prevent the entry of sizable objects. The location of exhaust grilles is far less critical than that of supply outlets because the air flows toward them from areas which are much larger than the openings, and there is no concentrated high-velocity stream at any appreciable distance away (see Fig 261). Exhaust grids are usually located in a side wall with the bottom of the opening at or near floor level. However, ceiling locations are recommended for bars, kitchens, dining rooms, etc., where warm air will gravitate to the ceiling level. Some ceiling outlets of special design combine the supply and return openings in a single unit. Low sidewall locations should not be placed closer than 5 ft to an area normally occupied by people who are seated. Exhaust grilles set in the floor will perform satisfactorily but are objectionable from the standpoint of dirt collection. Allowable velocities through exhaust grilles are largely dictated by pressure-loss and noise-level considerations. Gross-area velocities between 400 and 800 fpm are usually satisfactory from every standpoint.

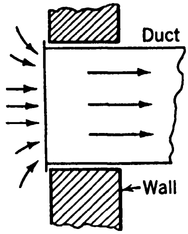


Fig. 261. Air entering an exhaust duct.

It is sometimes economical to use corridors as exhaust ducts. Air from the individual rooms passes into the corridors through grids set in the partitions or through an opening provided at the bottoms of the doors. Another scheme that is frequently used is to construct an exhaust duct at the top of a corridor by providing a false ceiling the required distance below the concrete floor above. Where such a duct is constructed without the use of metal the plastering of the ceiling must be very tight to prevent dirt streaks caused by air entering the duct from the corridor through small cracks. The same procedure may be used for designing an exhaust system as was explained in Art. 303 for a supply system. However, in estimating the friction loss of such a system, allowance should be made for additional friction due to disturbance of the main stream when branches are brought into an exhaust trunk from the side.

PROBLEMS

1. A mixture of dry air and water vapor exists at 14.5 psia, 80 F dbt, and 40 per cent relative humidity. Six-thousand pounds per hour of the mixture are to flow with a velocity of 20 fps in a duct having a rectangular cross section with a dimension ratio of 2 to 1. Find the size of the duct in inches.

2. Air-velocity determinations of 2.5 and 5 fps are obtained by use of a kata thermometer for air having temperatures of 68 F and 95 F, respectively. If the instrument factor *F* is 450 what cooling time is required in each case?

3. A pitot tube operates with a gage having its fluid at 60 F. Velocity-pressure determinations of 0.09, 0.64, 1.0, and 4.0 in. of water are made in air streams having a temperature of 90 F and a barometric pressure of 29.3 in. of mercury. What velocity of flow in feet per second is represented in each case?

4. If the velocity pressures of problem 3 were obtained with air having a density of 0.08 lb per cu ft would the rates of flow be greater or less than those existent for the conditions stated? Express as a percentage of the actual velocities.

5. The temperature of the fluid in a gage used with a pitot tube is at times 35, 70, and 130 F when air having a density of 0.075 lb per cu ft is handled. What velocity of air flow is represented in each case if the velocity pressure is held constant at 4 in. of water?

6. Air has a velocity of 2400 fpm when its density is 0.065 lb per cu ft and the pitot tube gage-fluid density is 62.4 lb per cu ft. What is the velocity pressure in inches of water?

7. An air-intake grille has gross dimensions of 30 by 42 in. and a free area of 80 per cent of its gross area. The average velocity of air flow at the grille face, as indicated by an anemometer, is 200.5 fpm. The instrument calibration indicates that the anemometer determination at the apparent velocity of flow is 2.5 fpm too high. What weight of dry air is passing through the grille each hour if its density is 0.075 lb per cu ft?

8. A fan has an inlet duct with a diameter of 68 in. and an outlet 49 by 64 in. The fan discharges 34,400 cfm of air measured at 75 F and an absolute pressure of 29.47 in. of mercury. The operating static pressures are −0.33 in. near the inlet and 1.25 in. of water at the outlet. Find the total-pressure difference which the fan creates.

9. A fan draws air through a short inlet duct in which the friction-pressure losses are 0.05 in. of water. The inlet-duct opening has a grille through the free area of which air flows at 20 fps with a pressure loss of 1.5 times the free-area velocity pressure. The air exists at 80 F and a barometric pressure of 28.95 in. of mercury. The flow conditions are: inlet duct, 30 fps; outlet opening, 40 fps; and discharge static pressure, 0.5 in. of water. Find the total-pressure difference created by the fan when the gage-fluid density is 62.3 lb per cu ft.

10. The number 2 Type-HV fan of Table 92 is operated at its maximum efficiency with a free inlet and 2 in. of water static pressure at its outlet. Compute the total and static efficiencies of the fan if the gage-fluid density is 62.4 lb per cu ft.

11. Select an HV-type fan to handle 17,330 cfm of air, measured at 115 F and a barometric pressure of 29.53 in. of mercury, when the discharge static pressure is 0.75 in. of water. State the required operating speed and horsepower.

12. A duct has a diameter of 48 in. and a length of 60 ft in addition to two 90-deg elbows which have centerline radii of 1.5 times the duct diameter. Air is handled at 1500 fpm with a temperature of 100 F and a pressure of 14.5 psia. Find the actual friction-pressure losses in inches of water.

13. Derive equation 133.

14. An 18 by 48 in. duct conveys 9000 cfm of dry air having a density of 0.071 lb per cu ft. Find the friction losses in 200 ft of equivalent duct length.

15. A circular duct of 18-in. diameter conveys air at 1800 fpm. Find the film conductance f_i.

16. A sound-intensity level of 40 db exists. What is the power of the sound waves?

CHAPTER 14

VENTILATION AND AIR PURIFICATION

313. Distinctions between Ventilation and Air Conditioning. Definitions of ventilation and air conditioning, Arts. 3 and 4, indicate that both have common features in the change and circulation of air within a space but that in true air conditioning much more air processing is often necessary than is required for ventilation. Outside air taken for ventilation purposes may or may not be heated in winter and have no treatment whatever in the summer. Ventilation systems employ either natural or mechanical forces to produce air flow and changes. Because of the amount of apparatus required and its resistances to air flow most air-conditioning systems have mechanical circulation of the air handled.

314. Necessity for Ventilation. Air changes in spaces may be required for any or all the following reasons: (1) heat and moisture given off by the occupants, (2) vitiation of the air by respiratory processes and emanations from the skin, (3) chemical product fumes and vapors, (4) oxygen depletion as the result of the combustion of fuels and other materials, (5) excessive heat from all sources, (6) moisture liberated in manufacturing and other operations, (7) toxic gases, (8) odors, (9) bacteria, (10) smoke and fog productions, and (11) dirt, dust lint, etc., either drawn from without or liberated within a space to float in its air.

Any appraisal of the functioning of a system of ventilation should give due consideration to its ability to eliminate or at least minimize the undesirable conditions that any or all the items of the foregoing list may produce. Certainly any satisfactory and successful system of ventilation should provide, within the space which it serves, air with adequate volume and circulation to give: sufficient oxygen; freedom from the dangers of toxic gases, vapors, and fumes; the reduction of odors to an acceptable threshold of intensity; and satisfactory conditions of air temperature, humidity, and cleanliness.

315. Air Vitiation. Heat and moisture losses from people in crowded spaces may lead to conditions far from comfortable. The respiratory processes of men reduce the oxygen and increase the carbon dioxide and moisture contents of air surrounding them and pollute it with organic materials from the nose, mouth, throat, and lungs. Skin emanations are another source of air pollution and odors may arise from them as well as from body uncleanliness, soiled and sweaty clothing, cosmetics, etc. Air odors come from many sources and irrespective of their origin they may or may not be indicative of harmful materials in the air. From a psychological standpoint they are objectionable and often lead the room occupant to feel that the ventilation of a space is poorer than it actually is.

Fresh air from the open country may have from 3 to 4 parts of carbon dioxide, CO_2, per 10,000 parts of air. Air as exhaled by an adult usually has about 400 parts of carbon dioxide per 10,000 parts of air. Concentrations of from 7 to 8 parts of CO_2 per 10,000 parts of air are not excessive and may be considered as satisfactory. With increased amounts of CO_2 an individual may suffer langor, drowsiness, a more rapid rate of respiration, headache, nausea, and with extreme concentrations insensibility may result. For the most part the usual concentrations of CO_2 as found in occupied spaces are not considered to be of great importance except as indices of air distribution and the adequacy of the ventilation system.

Carbon monoxide, even in very small quantities, is a deadly gas as far as human life is concerned. Moreover it is not noticeable to the individual inhaling it. Carbon monoxide is absorbed by portions of the blood so that proper assimilation of oxygen cannot be made. In the ventilation of garages, workshops, and vehicular tunnels, where quantities of CO are given off in the exhausts of internal-combustion engines, the amounts of ventilation air supplied must be such that the concentration of CO will not exceed one to four parts in 10,000 parts of air of the space. Many other toxic gases and vapors are possible air contaminants, some of which are also highly combustible and explosive in their nature.

Obnoxious and undesirable air impurities are dusts, dirt, plant pollens, and bacteria. Such contaminations arise from industrial processes, cinders, ashes, soot from chimneys, dust and dirt liberated from deposits on city streets, and organic materials including the fine powder from flowering plants known as pollen. These materials are of varying degrees of fineness, chemical activity, and injuriousness when inhaled. The dusts and various forms of dirt are undesirable from the standpoint of maintaining cleanliness of the structure and its furnishings.

The sizes of dust and fume particles are designated in terms of microns; a micron is one-millionth of a meter or approximately 1/25,000 of an inch. Dust is a solid material in a finely divided state, the particles of which have diameters ranging from 1 to 150 microns.

Fume particles arising from chemical actions range from 0.2 to 1 micron in diameter. Smoke in its different forms results from the incomplete combustion of different materials and generally carries particles less than 0.3 micron in size. Certain pollens are harmless; others are very injurious to individuals susceptible to them, causing the disagreeable malady known as hay fever. Bacteria range from the harmless to the dangerous.

Dust counts may be made by removing from a given quantity of air the suspended materials by settlement, impingement, scrubbing, and electrical precipitation. Bacteria determinations are usually made by exposing culture plates to the air for a definite time, 2 min., and then maintaining the cultures at a temperature of 98 F for a period of 24 to 48 hr to permit incubation. After the colonies are developed bacteria counts are made by use of a microscope or a magnifying glass.

316. Air Ionization. The ions of air are both positive and negative. Ionization of the air may possibly have some bearing upon its desirable qualities especially where a considerable portion of it is recirculated. Experiments have shown that there is a marked decrease in the number of both the positive and negative ions of the air of a space after a period of occupancy. The ionic content of air varies with temperature, occupancy, locality, and other conditions. The total effects of air ionization are indefinite and at the present are ignored.

317. Quantities of Ventilating Air Required. The amounts of air supplied for the purposes of ventilation may be fixed by: legal requirements; those necessary to serve as a carrier of either heat or moisture, or both; those necessary to produce satisfactory distribution and motion in occupied spaces; the quantities required to dilute carbon dioxide, odors, and toxic gases to unobjectionable concentrations; and the volumes used for the removal of floating dirt particles, smoke, dusts, and fogs. The air requirements for ventilation therefore are dependent upon local conditions and the ultimate results to be obtained.

When the source of air contamination within a space is its human occupant, factors which seem to affect the minimum quantity of ventilating air to be brought in from outdoors are: objectionable body odors as influenced by the personal and dietary habits as they are related to his social and economic status, the room-air temperature and relative humidity, tobacco smoke, the volume of the room per occupant, and the odor-absorbing capacity of the air conditioning processes. Data obtained from studies made by the Harvard School of Public Health[1] are included in Table 103.

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[1] ASHVE Research Report 1031; "Ventilation Requirements," by C. P. Yaglou, E. C. Riley, and D. J. Coggins, *ASHVE Trans.*, Vol. 42, 1936.

TABLE 103

MINIMUM OUTDOOR AIR REQUIREMENTS FOR VENTILATION UNDER VARIOUS CONDITIONS

Type of Occupants	Air Space per Person, Cu Ft	Outdoor Air Supply, Cfm per Person
Heating season with or without recirculation.	Air not conditioned	
Sedentary adults of average socio-economic status	100	25
	200	16
	300	12
	500	7
Laborers	200	23
Grade-school children of average class	100	29
	200	21
	300	17
	500	11
Grade-school children of poor class	200	38
Grade-school children of better class	200	18
Grade-school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person		
Sedentary adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person		
Sedentary adults	200	<4

The data of Table 103 are indicative of the amounts of fresh outside air necessary to reduce the intensity of body odors to a tolerable value. Attention is called to the fact that these amounts vary considerably depending upon the status of the occupants and the volume of room space available to them and that the extremes of volumes range from 7 to 38 cfm per person with possibly a satisfactory range between 10 and 30 cfm. Considerations involved in space ventilation are: the system of ventilation employed; the temperature, velocity, and relative humidity of the air handled; the locality; the climate of the locality; the physical features of the building; and the nature and duration of the occupancy. Residential spaces may have sufficient fresh air admitted to them through leakages through window or other cracks or by virtue of open windows. Crowded rooms are not well adapted to open-window sources of fresh air because of the likelihood of objectionable drafts; therefore in such cases some form of a mechanical system is desirable. For ventilation requirements and objectives to be attained in such work, reference may be made to The Code of Minimum Requirements for Comfort Air Conditioning.[2]

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[2] *ASHVE Trans.*, Vol. 44, 1938.

Because of the loads imposed on air-conditioning plants, especially in hot weather, by the use of large amounts of outside air in ventilation of spaces there is a tendency to reduce these amounts to a minimum of from 5 to 15 cfm per occupant except where smoking is prevalent and where larger quantities are necessary to carry away heat, moisture, dust clouds, and vapor fogs. The following data are of aid in estimating the amount of air required for ventilation purposes.

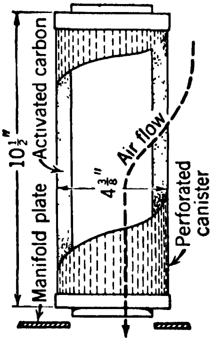
TABLE 104

MINIMUM NUMBER OF AIR CHANGES PER HOUR REQUIRED FOR VENTILATION*

Class of Service	Changes	Class of Service	Changes
Assembly halls	5 to 10	Laboratories	3 to 10
Bakeries	3	Offices	5 to 10
Billiard and pool rooms	3 to 8	Restaurant, dining room,	5 to 10
Bowling alley	3 to 8	kitchen	2 to 3
Dance halls	3 to 8	Stores	5 to 10
Foundries	5 to 10	Theaters	3 to 8
Garages	5	Toilets	3 to 5

* From "Air Conditioning and Engineering," by The American Blower Corporation.

318. Odor Suppression. Individuals when exposed to odors for some period of time become less conscious of them because of the dulling of their olfactory sense. In appraising odor conditions within a space attention should be given to the impressions of individuals entering the rooms and those of the room occupants. The data of Table 103 were secured by the method of dilution of room air with fresh air, and the results were based on the impressions of both room occupants and the primary impressions of investigators just entering the space.

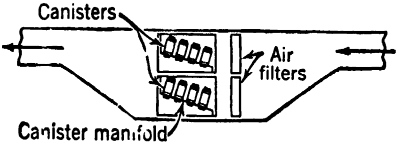


(*W. B. Connor Engineering Corp.*)

FIG. 262. Canister with activated carbon for odor adsorption.

Odor-bearing materials and vapors carried by air are successfully adsorbed by the use of activated carbon (charcoal made from coconut shells) placed in perforated canisters, Fig. 262, mounted on manifold plates. These assemblages, Fig. 262, are installed in duct sections, Fig. 263, through which the air to be deodorized flows. Whenever the activated carbon becomes ineffective as an adsorbent it may be regenerated by heating it to a temperature of approximately 1000 F. Each canister has an air capacity ranging from 25 to 35 cfm with air resistances of from 0.15

to 0.20 in. of water pressure. Activated-carbon requirements vary from 25 lb per 1000 cfm of air in residential applications to 35 lb per 1000 cfm in commercial installations. Filtration of the air entering odor-adsorbing units is advisable as their effectiveness is reduced by accumulations of dust on the activated carbon.



(*W. B. Connor Engineering Corp.*)

FIG. 263. Installation of odor-adsorbing equipment in an air duct.

Ozone, O_3, has been employed to mask odors as it has a pungent smell resembling that of weak chlorine. Because of its likelihood of causing distress and injury to humans its concentrations are limited to 0.01 to 0.05 ppm (parts per million) of air. With these low concentrations ozone is not effective either as a masking or a sterilizing agent.

319. Air Sterilization. Bacteria from human and other sources exist in occupied spaces. These germs may be harmless or otherwise. The distances that pathogenic germs travel are variable and are dependent upon both their method of introduction into the space and how they are carried about. Dusts, lints, and other materials floating in air afford a method of conveyance for bacteria, and the removal of such particles in air washers, filters, electrostatic cleaners, and other devices provides one method of at least partially freeing air from harmful germs.

Air sterilization by use of special ultraviolet-ray lamps which give radiations in the neighborhood of 2600 angstrom units[3] has shown results indicative that such equipment may be successfully used. Possible locations for ultraviolet-ray lamps are in the upper portions of rooms and in the air-supply ducts, just outside of the spaces. In any event lamps used for the purposes of air sterilization must be shielded so that the eyes and the skin of the occupants of the space will not be exposed to them.

Progress has been made in the use of small quantities of finely divided droplets or mists of either propylene or triethylene glycol sprayed into air for the purposes of killing bacteria. Effective air sterilization depends upon either the radiations of lamps or the mists of germicides being at all times capable of reaching the organism to be killed. Bacteria carried on air-borne particles of dust, dirt, lint, or other suspended material may be shielded from and only partially affected by any agent used for their destruction.

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[3] An angstrom is a measure of length, 250,000,000 of which equal one inch. As applied in radiation studies they refer to wave lengths.

320. Air-Cleaner Functions. Installations of air-cleaning devices are made for the purpose of removing from air, flowing in either an air-conditioning or a ventilating system, solid particles of materials carried with it. The natures and amounts of air-borne solids to be removed depend upon local conditions which involve sources of air pollution and winds including their directions and velocities. In general air cleaning includes the removal of various-sized particles of dusts, dirts, smoke, cinders, lints, pollens, and other materials.

An air cleaner should function to remove efficiently impurities even though the velocity of air flow varies over a considerable range, and it should offer small resistances to the flows. The cleaner requirements include large dust-holding capacity, ease of operation and servicing, and freedom from air contamination by the device. Neither by their construction nor by accumulations should air filters be fire hazards.

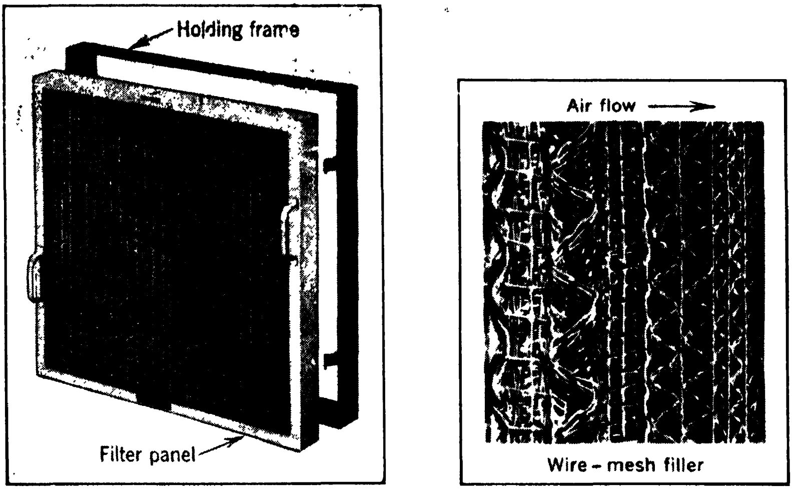
321. Air-Cleaner Classifications. The nature of the impurities to be removed and the thoroughness of air cleaning necessary, fix to a considerable degree the particular type of equipment to be used for given conditions. In general, cleaners may be separated into the following groups according to the principle of their operation: (1) filters with viscous-coated surfaces upon which dirt particles are impinged, (2) dry filters which consist of cellulose or cloth materials which screen out dirt particles, (3) devices that operate with a change of direction of the air flow so that dirt particles are thrown out of the air stream by the action of centrifugal forces, (4) electrical precipitation in which dirt particles are ionized and given a positive charge of electricity which allows them to be deposited upon surfaces or plates having a negative charge, and (5) air washers which by means of water sprays and the impingement of dirt materials upon wetted surfaces accomplish air cleaning and, dependent upon the conditions of operation, either humidification or dehumidification of the air handled.

Dirt and other materials collected by air cleaners must be removed from them, and the schemes employed are classed as (1) automatic and (2) non-automatic. Automatic rejuvenation of air cleaners will be discussed in connection with particular types of equipment illustrated. Non-automatic cleaners, when dirty, may be rehabilitated by (1) discarding (throw-away) dirty elements and replacing them with clean ones; (2) manually cleaning the dirty parts without removing them from place; and (3) by removing those parts in need of attention, renovating them, and replacing them in the apparatus.

Applications of air cleaners are in the following categories: (1) general uses which include central systems of air-conditioning and ventilation, unit ventilators, window installations, and warm-air furnaces;

(2) removal of objectionable particles from smoke and stack gases; and (3) the collection of dusts in exhaust systems. Non-automatic cleaners, according to the maximum resistance they offer to air flow in inches of water, are: low, 0.18; medium, 0.50; and high, one or more. The pressure losses through air filters are dependent upon the velocity of air flow and the state of cleanliness of the unit.

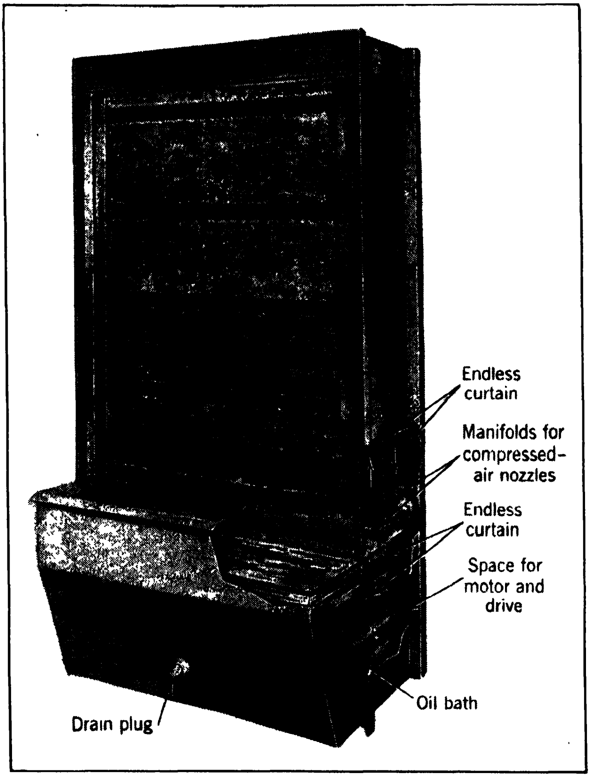
322. Viscous-Impingement Air Filters. Units of this type are built in several forms, one of which is the panel or cellular form illustrated in Fig. 264. The frame of the cell is sometimes of steel and other times of cardboard. The cardboard construction is for those units which are discarded after use has rendered them unfit for further service. The cells are filled with some material such as, steel shavings, vegetable fibers, spun-glass fibers, animal hair or bristles, and in the unit shown in Fig. 264 the filtering surfaces are formed by the use of crimped wire-mesh screens diagonally placed within the panel. The surfaces upon which dirt impingement takes place are coated with an oil. The properties of the light oil used are: (1) freedom from odor, (2) fire resistant, (3) non-volatility, (4) free flow at low temperatures, (5) slight changes of viscosity with temperature change, (6) ability to prevent growths of mold and bacteria on the filter medium, and (7) ability to make a uniform coating on the filter surfaces.



(*Air-Maze Corp.*)

FIG. 264. Viscous-impingement air-filter panel, holding frame, and wire-screen mesh.

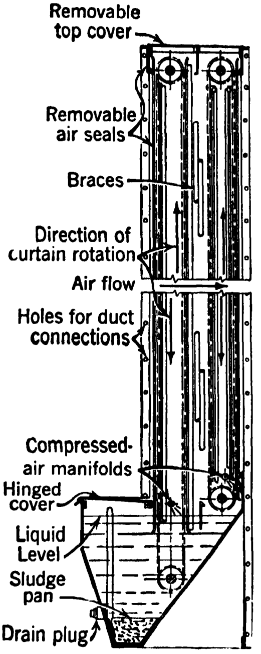
The cellular units assembled to form a barrier to dirt flow, Fig. 263, are installed between the source of dirty air and the activated-carbon deodorizing units. A common location is on the suction side of the fan. The assemblage of cells of the type of Fig. 264 can be such as to allow the passage of air without high velocities. The best results are obtained when the air can be divided into fine streams to impinge upon the filter material. When the filter agents are made of different sizes of fiberous materials or steel shavings the coarser elements are placed in the front of the cell where the larger particles of the dust are removed. Filters of this type continue to collect dust as long as the dirty surfaces have their dust particles oil coated. The need for cleaning the surfaces and recoating them with oil or disposal of the units occurs when the resistance to air flow becomes excessive.


Endless curtain
Manifolds for compressed-air nozzles
Endless curtain
Space for motor and drive
Oil bath
Drain plug

(*Dollinger Corp.*)

Fig. 265. Sectional view of an automatic viscous air filter.

Automatic viscous filters have moving surfaces which are generally periodically moved. Figure 265 is the front view of an automatic viscous unit with two endless movable filter curtains, Fig. 266, which are mounted on roller chains driven by sprocket wheels keyed to the shafts of the curtain rollers. The curtains have a number of removable panels each constructed of a single layer of bronze screen wire to which are attached layers of woven copper mesh. Lint is caught on the bronze screen wire which is always on the air-entering side of the curtain. The inner mesh affords a considerable surface for the collection of dust particles. The first curtain, Fig. 266, is the denser of the two and dips in an oil bath through which it passes for cleaning. The curtain at the exit side of the unit does not enter the oil bath but serves to prevent entrained oil from passing with the air from the cleaner. The front part of the curtain at the air inlet moves downward into the oil bath, and the air after the first contact with the curtain passes over freshly cleaned surfaces. A solenoid valve operates compressed-air jets working at 1½ psig pressure to blow through the curtain panels just below where they enter the main air stream to remove the excess oil from the clean surfaces.



(*Dollinger Corp.*)

FIG. 266. Side sectional elevation of automatic viscous air filter.

323. Dry Filters. These cleaners depend upon the screening or straining action of the filtering material, which may consist of felt cloth, wire screens, or cellulose. No adhesive liquid is used to coat the filtering surfaces. The filtering materials are often arranged in pocket forms to give the necessary surface in a small space. The separate units are so placed in multiple to give the required air-passing capacity. When either cellulose or cloth is the filtering material, it may be cleaned either by vibration of the material, dry cleaning of cloth, or by vacuum cleaning, or cellulose filters may be discarded when dirty. The velocities of air flow permissible range from 18 fpm for felt to 35 fpm for cellulose. Excessive resistance must not be allowed to exist, and dry filters must be kept free from moisture.

324. Centrifugal Separation and Dynamic Precipitation of Dust Particles. When the particles of materials carried by air are large, such as sawdust and plane shavings, mechanical separation of the two can be secured within an inverted cone-shaped chamber such as shown by Fig. 267. Here the action of centrifugal forces and the reversal of direction of air flow as it escapes from the unit effects the removal of the air-borne solids.

A piece of equipment which makes use of centrifugal separation and dynamic precipitation of dusts and dirts is that shown by Fig. 268. This particular unit consists of a scroll-shaped casing within which a specially designed turbine-type impeller wheel, with many blades, operates at the speed of a conventional electric motor. Under centrifugal action dirt-laden air entering the wheel at its center flows outward along the wheel blades to enter the discharge scroll at a point where the width of the impellers is materially reduced. Impingement of the heavier dirt particles occurs on the impeller base, and the lighter particles are deposited or precipitated on the advancing blades. The actions of centrifugal forces move both the light and the heavy particles to the outer edge of the impeller where their paths converge and intercept each other because of the hyperboloid blade contour.

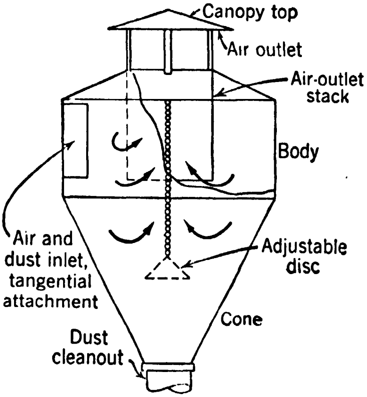
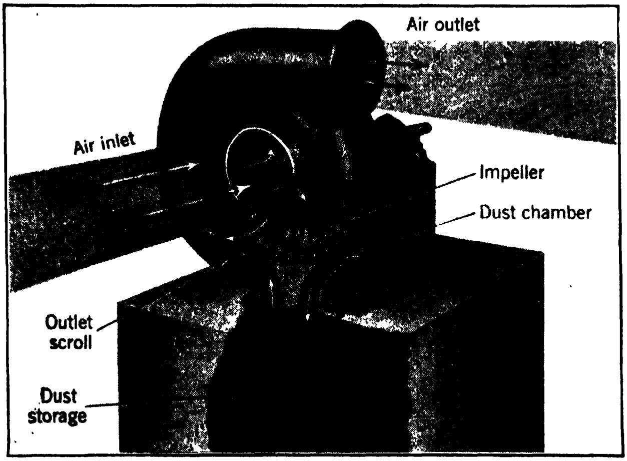


FIG. 267. Cyclone dust collector.



(American Air Filter Co.)

FIG. 268. Roto-Clone air cleaner.

About the periphery of the wheel, directly opposite the discharge from the blade tips, is a narrow annular space through which the separated dirt passes into a surrounding dust chamber. The blade tips extend into the dirt chamber and produce a secondary-air circulation which in turn conveys the dirt and dust particles through a port whereby the materials collected are discharged to a storage hopper or other receptacle. The cleaned air leaving the blades and passing out into the scroll passage is discharged either to the outdoors or is returned to the occupied space after having been passed through an air filter. The air which leaves the blade tips and flows with the dirt into the collecting receptacle passes back to the dust chamber at the wheel periphery. This air is recirculated as air is not bled from the main stream of air discharge for secondary flow.

This process of air cleaning is used principally in industrial plants where dangerous and noxious dusts are not to be liberated in the community. The process thus far described is a dry one, as water is not used. Equipment of a similar type is available whereby water sprays are used to effectively wet dust and dirt particles and thereby increase their masses which is of material aid in separating them from air. The wetted surfaces within the precipitator also show increased resistance to wear and abrasion of the dirt particles.

325. Air Cleaning by Electrical Precipitation. Fine dusts, unburned particles in tobacco smoke, and other minutely divided substances may pass through either dry or viscous air filters. Electrostatic cleaners are quite effective in removing small particles that some other devices cannot separate from air.

Fundamentally the principles of the electrical precipitator are as explained by the use of Fig. 269, although the actual construction details of commercial units vary with different designs. Generally the source of electrical energy for their operation is 110- to 115-volt alternating current of either 25 or 60 cycles frequency and single phase. By means of electronic devices in a power pack a transformation is made to direct current which is available at potentials of 12,000 and 6000 volts.

The ionizing wire *I* is energized at 12,000 volts and thereby sets up an electrostatic field between it and the ground tubes *G* and *G′*. These tubes have grounded plates, shown by the solid lines of Fig. 269, attached to them. The polarity of both the tubes and the attached plates is minus or negative. Positively charged plates, shown by the dotted lines, are spaced between the negatively charged ones and are connected to the 6000-volt direct-current terminals of the power pack. Dirt, mists, lints, and vapors flowing with the air through the field between *I* and *G* and *G′* pick up electrical charges as the air becomes ionized. Some manufacturers of this type of equipment state that all the air-borne particles acquire a positive charge of electricity, while others place the process as having given more than 80 per cent of the bodies a positive charge and less than 20 per cent a negative charge. In any event the charged particles pass between plates having positive and negative electrical charges, and they are deposited upon a surface having a charge opposite in sign to that which they carry. The path of a positively charged particle is shown in Fig. 269.

Westinghouse Precipitron cells are composed of two duplicate sets of high-voltage and grounded plates with the ionizer as a separate unit to facilitate installation and servicing. The cell sizes are listed as 36 in. by 24 in. and 24 in. by 24 in. at the face where the air enters them. Operating face velocities are 300 and 375 fpm, which give capacities of 1800, 1200, 2250, and 1500 cfm respectively with cleaning efficiencies of 90 per cent at 300 fpm and 85 per cent at 375 fpm velocity. These cells are installed in sufficient numbers of a given size within the duct work of central systems to give the necessary air capacity when the operating face-area velocity is fixed. Cleaning of the accumulations on the grounded plates is accomplished by use of water discharged from a hose nozzle to flush away the collected materials. Raytheon units are built in the same sizes and capacities as the Precipitron cells and are used for central installations as well as smaller room units.

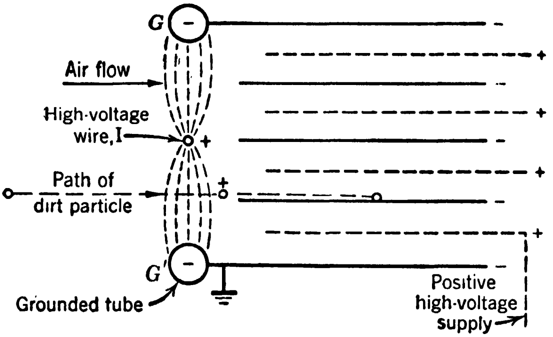
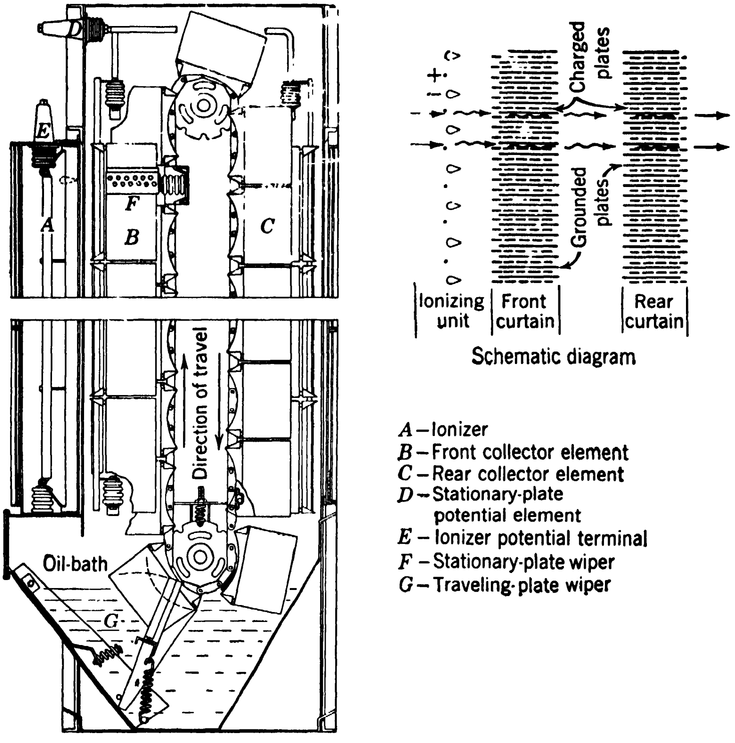


FIG. 269. Principle of electrostatic air cleaner.

A self-cleaning type of electrical precipitator is shown together with its schematic diagram by Fig. 270. In operation the dirty air first passes through the ionizer with its positive high-voltage wires and its negative and grounded tubes. The potential difference between the ionizing wires and the grounded tubes is approximately 12,000 volts, and in this region the polarity of the ions or the dust particles is established. The collector elements consist of fixed positively charged and movable negatively charged plates. The ionized air-borne particles enter the field between the fixed and the movable plates and are drawn to the surface of a plate having a polarity opposite to theirs. Particles not removed in the front collectors pass to the rear sections where there is further opportunity for their removal. The negative collector plates in both the front and the rear locations are moved through a bath of oil placed at the bottom of the assembled unit. As the plates move through the oil they come in contact with a wiper which removes tarry residues which have formed on the surface of the oil film in contact with the air being cleaned. Dirt and other materials deposited upon the positively charged stationary plates are swept away from them by wipers attached to the moving chain on which


D
E
A
F
B
C
Direction of travel
Oil-bath
G
Charged plates
Grounded plates
Ionizing unit
Front curtain
Rear curtain
Schematic diagram
A–Ionizer
B–Front collector element
C–Rear collector element
D–Stationary-plate potential element
E–Ionizer potential terminal
F–Stationary-plate wiper
G–Traveling-plate wiper

(*American Air Filter Co., Inc.*)

FIG. 270. Cross-sectional and schematic drawing of an automatic electromatic precipitator air cleaner.

they are mounted. The speed of movement of the curtain of collector plates may be varied from one revolution per 24 hr to $\frac{1}{60}$ of its height in 12 min. The efficiencies of operation are 85 per cent at 500 fpm and 90 per cent at 400 fpm face-area velocity with air-flow resistances of 0.23 and 0.17 in. of water through the apparatus. These units are built in section widths of 3 and 4 ft and heights ranging from 6 to 13 ft. The necessary face area required for a definite velocity of flow is secured by the installation of units side by side.

The same manufacturer produces precipitators with stationary collector plates which may be removed from the assembly for their cleaning. Also the same builder makes a precipitator wherein sheets of cellulose material are spread over insulated grilles that contain both positive and grounded electrodes. These give a dielectric field in the cellulose material whereby the tissue fibers tend to separate and become dirt-collecting electrodes. The cellulose materials also serve as ordinary dry air-filter sheets and are discarded as they become dirty when used either as a plain filter or as part of an electrostatic cleaner.

326. Air-Cleaner Ratings.—The air flow in cubic feet per minute for which a cleaner is designed is an expression of its capacity rating. The dust-holding capacities of non-automatic filters are the amounts of dust that filters can accumulate with the resistances to air flow not in excess of some fixed amount as indicated in Art. 321. Cleaner ratings are usually determined when they operate with face-area air velocities somewhere between 250 and 500 fpm. Face-area velocities per unit of time are equal to the quantity of air flow through the filter, per unit of time, divided by the cross-sectional area of the duct at its point of attachment to the inlet side of the filter or the actual face area of the filter when the duct cross-sectional area is not the same.

327. Cleaning Efficiency of Air Filters. Much work has been done by various investigators in the matter of testing and rating air filters. An ASHVE Code[4] provides instructions in the matter.

The nature and the size of dust particles have much to do with their removal from air by filters. Coarse dirt particles are much more easily removed from air than those of fine size. Results of filter tests can best be compared when made with standard dust.

Filter cleaning efficiency or dust arrestment is expressed as

$$E = \frac{(M_1 - M_2)100}{M_1} \tag{136}$$

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[4] "Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work," *ASHVE Trans.*, Vol. 39, 1933.

where E = filter efficiency or dust arrestment, per cent.
M_1 = amount of dust or dirt in a unit volume of air entering the filter.
M_2 = amount of dust or dirt in a unit volume of air leaving the filter.

Cleaner efficiencies may be evaluated by means of (1) particle counts, (2) weights of dust introduced into air and the extraction of dust from cleaned air by passing it through a porous crucible, and (3) the dust-spot or blackness test as devised by the National Bureau of Standards.

Particle counts per unit volume of air are most useful in appraising filter performances with such materials as plant pollens and certain types of industrial dusts. The determination of the values of M_1 and M_2 by the weight method, indicated by item 2 of the preceding paragraph, is satisfactory and is recommended by the ASHVE. The dust-spot method requires the passing of cleaned and uncleaned air through filter papers at the same time. By adjustments of the ratio of the areas of filter paper through which air samples flow and the proper ratio of air quantities handled, spots of approximately equal blackness can be obtained on the papers. The ratios of the blackened areas and the ratios of the air volumes are indicators of the cleaners effectiveness. A special photometer is required for the comparison of the blackened areas.

The sizes of dust particles and their nature have a marked bearing on the performance of an air filter at a given rate of flow. Tests[5] have been made to determine the overloading of air filters by the introduction of various amounts of dust (50 per cent Pocahontas coal ash, 20 per cent lampblack, 20 per cent Illinois fly ash, and 10 per cent fuller's earth) into the air stream passing through the test unit. The dust was fed at the rates of 20 and 40 grams per hr to air flowing through a viscous-coated throw-away-type filter 2 in. thick. The curly fiber filter media were graded in size, density, and oiling, from the entering to the leaving side.

In the tests a filter was assumed to be overloaded when its efficiency of dust removal showed a definite and marked decline, as the feed at a given rate continues after a period of time. The curves of Fig. 271 give no definite indication of overloading of the filter, as the efficiency curves for each condition of dust feed show a rise in numerical values as the total amount of dust feed is increased. Naturally the filter resistance to air flow increased after a definite period of its operation with each rate of dirt feed. The rate of increase for both feeds was practically the same until the resistance to flow approached 0.3 in. of water after which the resistance to air flow became more rapid with the greater rate of feed when a definite amount of dust had been fed. The curves are of value as an indication of the efficiencies of operation and the resistances to air flow that may be obtained with one type of air filter.

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[5] "Overloading of Viscous Air Filters during Accelerated Tests," by Frank B. Rowley and Richard C. Jordan, *ASHVE Trans.*, Vol. 48, 1942.

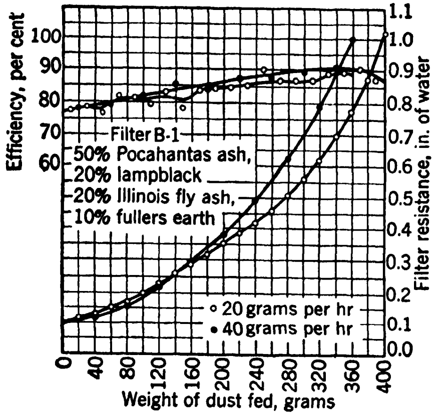
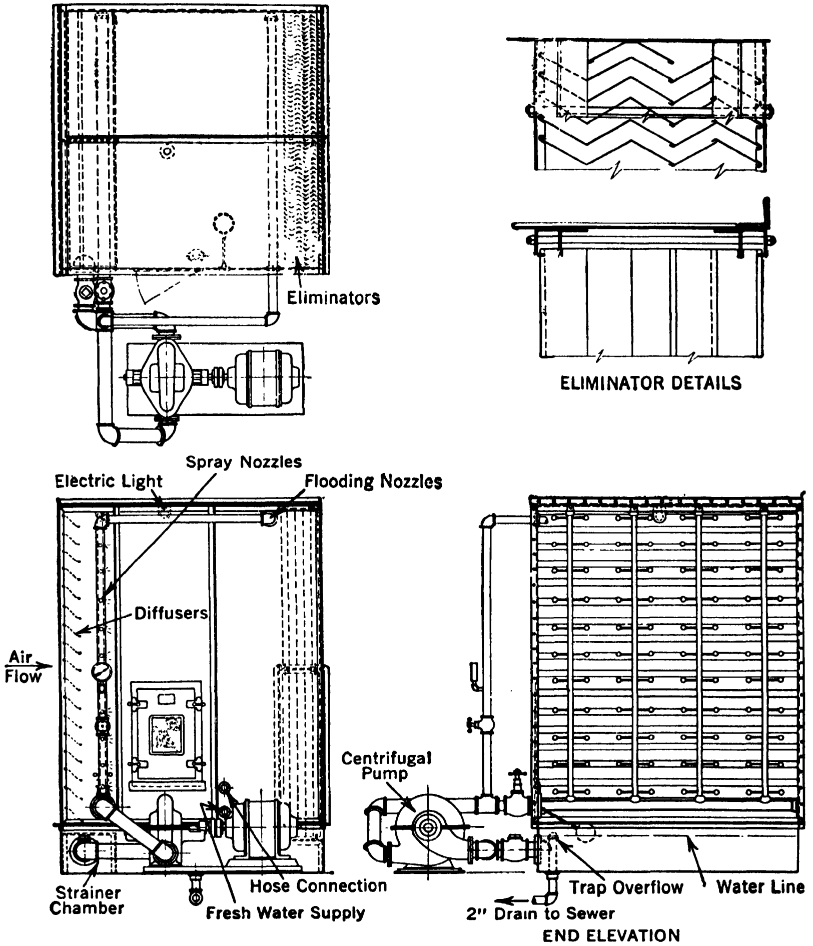


FIG. 271. Performance curves for a viscous air filter with two rates of dust feed.

328. Air Washers. Air-washing equipment serves as an air cleaner; it also functions as either an air humidifier or an air dehumidifier, depending upon the conditions of operation. Air washers remove certain dusts and dirts very effectively; other impurities such as fine soot are not readily removed from air by washing. This form of equipment functions by bringing the air to be conditioned in contact with (1) water broken into fine droplets as in a mist, (2) a combination of water-wetted surfaces and water discharged from spray nozzles, and (3) surfaces continuously wetted by water.

The air washer of Fig. 272 operates under the conditions enumerated by item 2 of the preceding paragraph. The assembled device consists of a spray chamber fitted with one bank of water nozzles, scrubber plates, and water eliminators at the air outlet all housed within the same casing. The diffusers at the air-inlet end give more uniform distribution of the air across the spray-chamber section. The action of the washer is secured by having the air pass through finely divided water sprays produced by nozzles placed in the path of the flowing air. The water is withdrawn from the sump or tank of the spray chamber and is forced through the nozzles by a pump usually operating with a discharge pressure of around 25 psig. Water not absorbed by the air in passing through the sprays falls to the sump where a constant level is maintained by the action of a float valve in the make-up water supply line. The sump is also fitted with an overflow to permit the escape of excess water. Figure 273 shows the application of a single-spray-bank washer in an air-conditioning system.

Washers similar to the one of Fig. 272 are built with two banks of spray nozzles, and they may have three banks within a single casing. Where exacting conditions are to be maintained washers may be arranged in series or tandem. Whenever the outside air temperature falls below 32 F a preheater or air-tempering coil must be installed in the duct just ahead of the washer air inlet. Unless the air entering the washer is maintained at a temperature above 32 F, say 40 F, the spray water will be frozen. The number and the design of spray



(*Westinghouse Electric Corp., Sturtevant Division*)

Fig. 272. Air washer.

nozzles are of great importance in air-washer operation. The water should be broken into a fine spray by the nozzles, and the contact of the air with the water sprays must be thorough. The velocity of the air in the spray chamber should be limited to 500 to 550 fpm.

Most of the air cleaning is done as the dust and dirt impinge on the scrubber plates which are flooded with spray water. The scrubber surfaces are the parts of the eliminator assembly with which the exit air first comes in contact. The eliminator plates which remove particles of entrained water from the air are also of aid in dirt removal. Change of the direction of air flow in the zigzag paths through the scrubbers and eliminators is of some value in the removal of dirt.

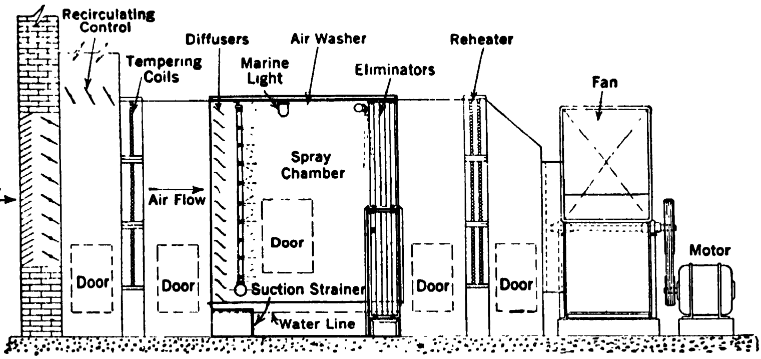
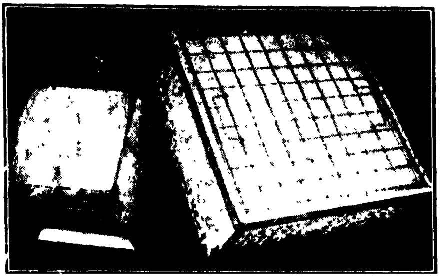


FIG. 273. Air washer application.

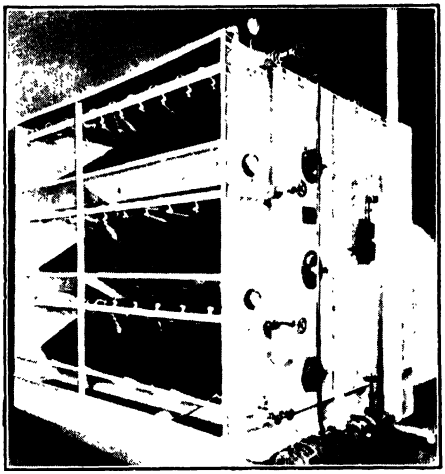
Air washers require frequent changes of water and the removal of accumulated dirt in the sump or spray chamber tank. Unless the spray water is changed frequently it may become saturated with odor-bearing materials and thereby be a source of odors in the air supplied to ventilated and air-conditioned spaces.

329. Cellular Air Washers. The basic feature of cellular air washers or conditioners is the employment of water-sprayed capillary cells, Fig. 274, through which the air to be processed flows. The unit of Figs. 275 and 276 has standard cells with face dimensions 20 in. by 20 in. and a depth of 8 in. in the direction of air flow. Each cell is completely filled with fine glass filaments which occupy about $1\frac{1}{2}$ per cent of its volume. A thin layer of the filaments is placed parallel to each cell face where the air enters and leaves the unit. The remainder of the filaments are placed practically parallel to the direction of air flow. The horizontal placement of filaments at the entering face aids in distributing water in the cell, and this same arrangement at the air exit from the cell facilitates the separation of water from the leaving air, Parallel arrangement of the filaments in the main body of the cell reduces resistances to flow and permits small moving streams of air and water to make maximum contacts with excellent heat and moisture exchanges.



(*Air and Refrigeration Corp.*)

FIG. 274. Construction of fiber-filled air-washer cells.

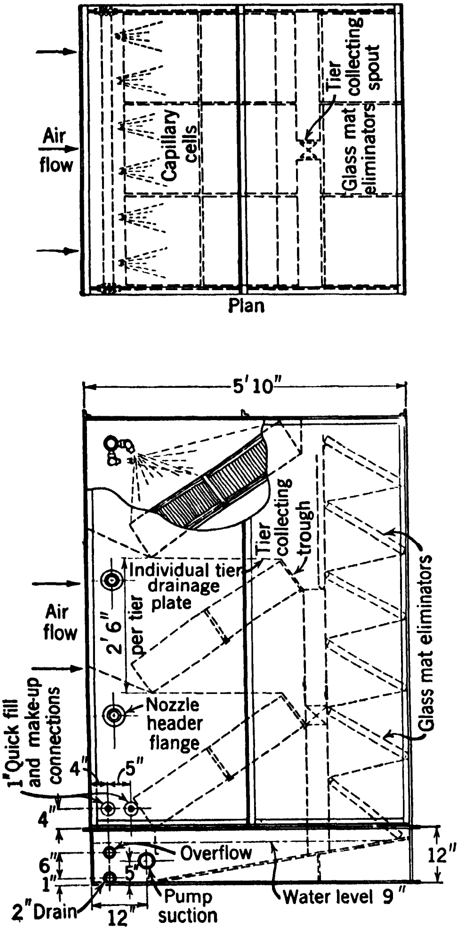


(*Air and Refrigeration Corp.*)

FIG. 275. Standard size 6-5 class I capillary central-station air-washer installation.

Each cell unit of this particular design is capable of handling 1100 cfm of air, but a nominal rating for all conditions of installation is 1000 cfm, which gives a face-area velocity of approximately 350 fpm. Conditioner units are assembled with the cells in one or more tiers and with a sufficient number of cells to give the desired width. The tiers are inclined to the vertical, and the washer illustrated by Figs. 275 and 276 has two horizontal rows of sections per tier. The washer shown by Fig. 275 has a 6-5 arrangement which means that there are three tiers of sections, each two sections high, and that the washer has a width of five sections or cells. Each tier operates independently of all others with respect to the air and water handled. An assembled unit, Fig. 276, includes a water sump, capillary cells, spray nozzles, circulating pump, water piping, water-level control, drain and overflow connections, and eliminators.

Water from nozzles above the cells is evenly distributed over their upper faces and passes by gravity flow through them to the bottom tank. Air flow may be parallel to the water flow or they may move counterwise when air enters the bottoms of the cells, In either case glass-fiber mats placed across the air exit or simple shallow metal eliminator plates function to remove the entrained moisture. Each cell tier is fitted with a trough and attached drain pipe which together return the excess water to the sump, The water pressure at the nozzles is approximately 6 psig. For ordinary washing, humidification, and evaporative cooling of air the amount of water pumped is about 3 gpm per cell or each 1000 cfm of air. When air cooling and dehumidification are affected the amount of water necessary is dependent upon the desired final air temperature, the initial temperature of the spray water, and the wet-bulb temperature of the entering air. The maximum amount of spray water to be handled has a recommended limitation of 9 gpm per cell.



Plan

Side elevation

(*Air and Refrigeration Corp.*)

FIG. 276. Details of capillary air washer.

Resistances to air flow through capillary washers are dependent upon the velocity of air flow or the quantity of air handled per cell and the amount of water used in gallons per minute per cell. Thus for 1000 cfm air capacity per cell the resistance to air flow when 3 gpm of water is used per cell is 0.34 in. of water; the pressure loss or resistance increases to 0.39 in. when the quantity of water used is 9 gpm. The saturating efficiency (humidifying efficiency) ranges from 99.5 per cent with 600 cfm of air per cell down to 96.9 per cent when 1400 cfm are flowing.

330. Systems of Ventilation. Ventilation is accomplished by natural and mechanical means. Natural systems comprise open windows, roof ventilators, and also vertical ducts which depend either upon wind action or gravity flow of air. Natural systems of ventilation cannot be used where air has to be passed through washers, filters, and many types of heating, cooling, humidifying, and dehumidifying equipment. Mechanical systems embody either central plants with distributing ducts or unit ventilators, with little or no ductwork, placed in the rooms of a structure. All mechanical systems require fans to produce positive flow of air.

331. Natural Ventilation. Systems of natural ventilation lack uniform positiveness in action as they are greatly affected by wind and temperature differences of the air. They have further limitations because of the size and nature of the ducts required.

Open windows and skylights are of value when weather and building conditions permit their use for ventilation purposes. The chief difficulty with open-window ventilation is the inability to prevent drafts and to secure uniform air distribution and air temperatures in occupied zones.

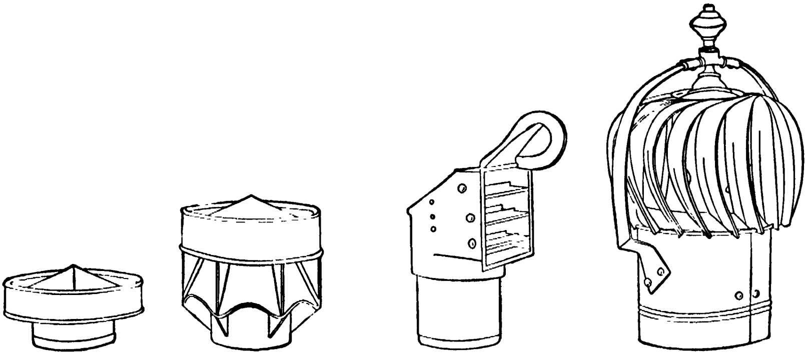
Special roof ventilators are also greatly affected by wind and weather conditions as well as by their height above the level to be ventilated and the inside-outside-air temperature difference. Roof ventilators are of considerable value in some types of factory buildings and for venting of heated air from attic spaces.

Other systems of natural ventilation embody special inlets, exhaust ducts, and outlets for the removal of vitiated air. The flow of air in ducts of natural systems is promoted by the difference in weight of two columns of air. Therefore the temperature differences and the height of the columns of air between a duct inlet and its discharge outlet are highly important. Exhaust ducts of natural systems are often provided with heating coils (steam or hot-water) which function to increase the motive head by increasing the average air temperature within the duct. In any system of natural ventilation the effects of wind and gravity action should work together rather than in opposition.

332. Roof Ventilators. These units are constructed in a variety of forms, four of which are shown by Fig. 277. The most desirable roof ventilators function to prevent the entry of rain and snow into the air outlet, permit the outflow of air regardless of wind direction and velocity, and when possible act in conjunction with wind forces.

333. Proportioning Natural Ventilation Stacks. The action of a stack in a natural system of ventilation is similar to that of a chimney. Equation 76 can be used for the purpose of arriving at the cross-sectional area of a gravity-flow stack, provided that the proper constants are used for K. When equation 76 is used, Q represents the cubic feet of air handled per second at t_c the average temperature of air in the stack, when t_a is the temperature of the outside air, and h the height in feet of the stack outlet opening above its inlet opening. Values for K are 0.35 for stacks having grilled inlet and outlet openings and 0.45 for those with free inlets.

334. Central Ventilation Systems. These systems having one or more fans operate: (1) to blow air through ducts into spaces with the air leaving the room through vent stacks or other openings, (2) to exhaust air through a duct system with the fresh supply of air entering at inlet grilles in the walls or window openings, and (3) both to blow air into and to exhaust air from spaces to be ventilated or air conditioned. In any event, if the air inlets and outlets are properly designed and placed, positive circulation of the air can be attained with a reasonable uniformity of distribution. Uniformity of distribution is of the greatest importance in ventilating and air-conditioning systems. Any system which merely blows a given quantity of air into a space and which exhausts a similar quantity without uniform distribution of the air in occupied zones falls far short of one of its essential purposes.



Plain Stationary
Stationary Siphoning
Rotary
Rotary Siphoning

FIG. 277. Roof ventilators.

Central supply systems of ventilation and air conditioning include those termed split systems and combination systems. The combination arrangements are further classified as divided and hot-blast systems.

335. Split Systems of Heating and Ventilation. In the split system the heat losses from the building are supplied by either direct radiators or convectors. Either the ventilation or treated air is handled by a fan and duct plant which delivers tempered air at the room temperature. This arrangement lends itself to various classes of buildings where ventilation is not required continuously. The operation of a ventilating plant in such buildings, in order to supply heat, would be expensive. The use of direct radiators and convectors permits spaces to be brought to and maintained at the desired temperature during periods of unoccupancy and also when positive ventilation is not a necessity. The ventilating part of the equipment can be arranged for complete air conditioning for either winter or summer conditions. This additional equipment for processing air to give complete air conditioning may involve air filters, air washers, and cooling coils.

336. Combination Systems. Conditions often arise where it is not possible or desirable to install a sufficient amount of direct radiators and convectors to care for the building heat losses. Sometimes better distribution of the heat within a space may be secured by having part or all the heat supplied carried by the air brought into the room from the central ventilating and air-conditioning plant.

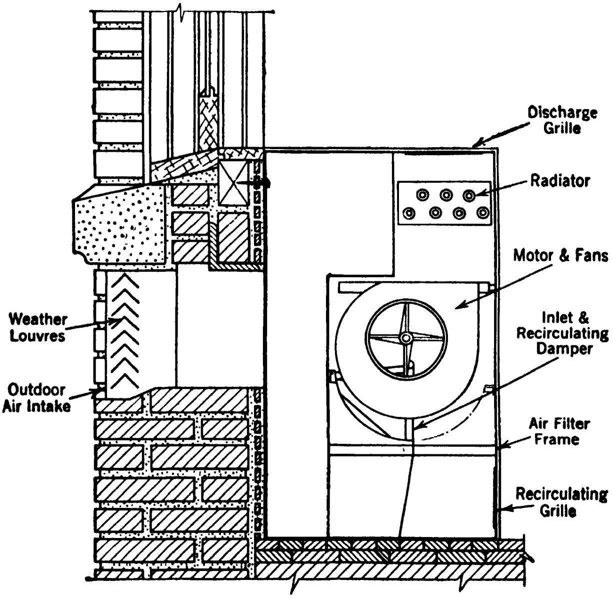
Where exposed glass areas occur, direct radiators placed beneath them are of considerable benefit in counteracting cold drafts produced by infiltration and also in offsetting the effects of the cold window surfaces. The divided system functions with part of the building heat losses supplied by either direct radiators or convectors, the remainder of the heat being brought in by ventilating air having a temperature higher than that of the room.

Where all the air supplied to the room is heated by the steam or hot-water coils of the ventilating system and no heater units are placed in the room the term hot-blast heating is applied. The temperature of the incoming air in hot-blast systems is adjusted according to the heating requirements, the quantity of air usually remaining constant as established by the ventilation requirements.

337. Unit Ventilators. Central systems of ventilation require considerable space in the building and its walls for the fans, heaters, and ducts. This necessity adds considerably to the first costs both of the structure and of the ventilating system.

With unit systems one or more small pieces of equipment are located within the space to be ventilated. A typical unit for both heating and ventilation, Fig. 278, consists of a housing, motor-driven fans, and a steam coil for warming or heating the air handled. It is often desirable to have the unit ventilator merely heat the air to room temperature or slightly less and to use direct radiation and convectors for the space heating. Unit ventilators are installed to handle outside air taken through short connections made through the building outside wall along which they are placed. Arrangement for the recirculation of inside air is highly desirable as the operation of the unit with recirculated air is of great aid in warming up a cold room. The vitiated air from the room can be disposed of through a system of vent flues, through a partially opened widow, or by outward leakage. Practically all unit ventilators embody either an air filter of the dry type or a cellular viscous filter of the throw-away type. Units of this sort are also available, which in reality are unit air conditioners embodying within a single housing the necessary features to clean, heat, humidify, and also to clean, cool, and dehumidify air.

Unit ventilators and air conditioners are particularly useful in schoolrooms, in connection with direct radiators and convectors. The air discharge at the top outlet is adjusted to discharge the fresh air at an angle with the room ceiling. By this arrangement drafts can be avoided and turbulence created in the air above the occupied zone. This leads to gentle circulation and better air distribution about the occupants of the room.



SECTION
(*Westinghouse Electric Corp., Sturtevant Division*)

FIG. 278. Unit ventilator.

338. Systems of Air Distribution. Central ventilation systems may be further classified according to the direction of air flow from the air inlets into the room, as upward, downward, and ejector systems.

339. Upward Flow Systems. The ventilating air supply in upward systems of ventilation, Fig. 279, is brought into the space through inlets near the floor line, through openings in the floor, or through the pedestals of theater chairs and seats. The foul-air outlets are located in the side walls near the ceiling or in the ceiling. Upward systems of air flow may be used in rooms where there is a marked tendency for air heated by the occupants to rise and carry with it the vitiating products from their bodies. These systems involve difficulties in the introduction of the ventilating air so that the occupants are not subjected to drafts and in the tendency to make the heating of the space more difficult. Upward systems of ventilation are not satisfactory where air cooler than the room temperature is to be introduced.

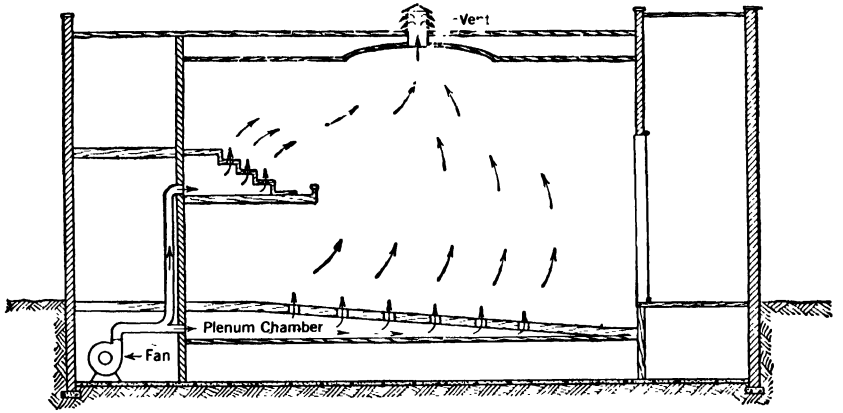


FIG. 279. Upward system of ventilation.

340. Downward Systems of Air Flow. The downward system of ventilation, Fig. 280, introduces the ventilating air through openings located in the ceiling and removes the vitiated air through openings either in or near the floor. The supply openings can also be located in the side walls above the heads of the occupants, and the vents in the side walls at the floor level, preferably on the same side of the room as the supply openings. Any system of downward ventilation should endeavor to spread the incoming air uniformly above the occupied zone to secure uniform conditions within the ventilated space. Downward systems of ventilation if properly installed are satisfactory for theaters, auditoriums, schools, etc. For summer cooling the downward system distribution is much better than the upward system.

Entering air cooled more than 10 to 15 deg F below the room temperature may be somewhat difficult to handle without objectional drafts because of its greater density than that of the air of the room. The downward system with properly designed diffusers located in the ceiling permits the rather uniform distribution of cooling air and tends to eliminate objectionable drafts.

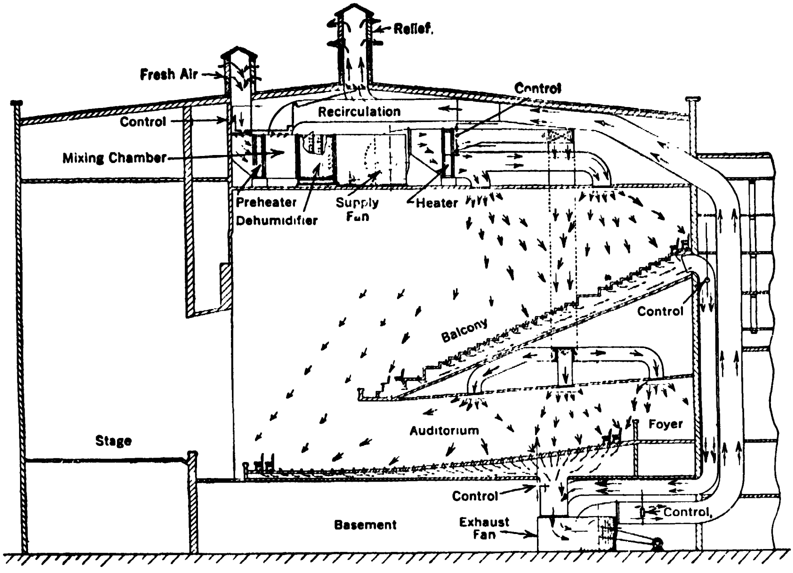


FIG. 280. Downward system of ventilation.

341. Ejector Systems. The ejector system of air distribution which is a form of downward distribution is feasible and satisfactory where the air may be blown through unobstructed spaces for considerable distances, as in Fig. 281. The air is introduced at high velocity through specially designed nozzles at the rear of the building and is discharged well above the occupied zone. The jet action of the

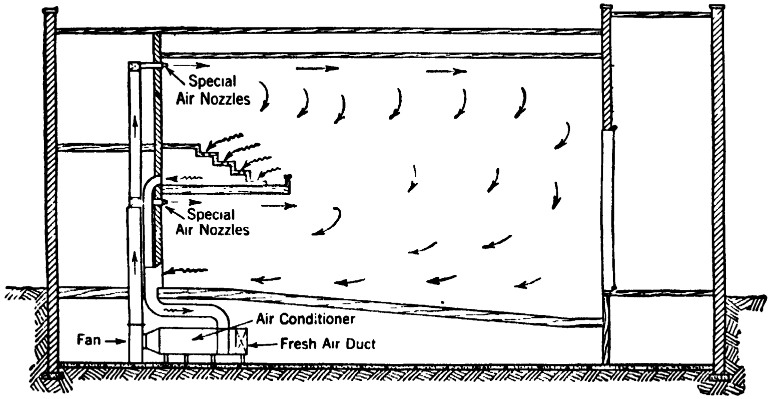


FIG. 281. Ejector system of ventilation.

incoming air produces considerable aspirating effects giving better diffusion of the fresh air within the space. The vent outlets are placed in the rear of the building, located so that the return air flows uniformly into the faces of the audience.

342. Air Exhaust. With any system of fresh-air distribution whatever there should be a positive scheme of vitiated-air removal so designed and located as to aid in the distribution of the fresh air and to prevent short circuiting of the air between the supply and the vent outlets.

When the air is supplied to maintain a pressure within the space at a value slightly greater than atmospheric pressure, leakage outward through cracks, etc., may be sufficient to care for the disposal of vitiated air.

A better method is to provide a system of vent flues or ducts through which the air can escape. These ducts may have air flow produced by either gravity action, air pressure in the room, or the action of an exhaust fan. Exhaust ducts from the various rooms may discharge the vitiated air to an attic space from which it may be exhausted by a fan, or there may be an exhaust fan and a trunk system of ducts having flue connections to the individual rooms. An exhaust fan attached to a trunk duct system to withdraw air through it gives the most positive results.

Exhaust vents operating with gravity action are designed to carry the same amount of air as is delivered to the rooms, and the air velocities in such ducts range from 200 to 600 fpm. Where fans are used in connection with exhaust systems, allowable velocities in the risers and main ducts of public buildings range from 500 to 600 and from 900 to 1500 fpm, respectively. For industrial buildings these velocities may be for risers 900 to 1800 and in main ducts 1200 to 2400 fpm. Gravity flow in exhaust systems is always more or less erratic in action, and if positive air removal is desired or if the exhaust duct system is of considerable length a fan should always be used in connection with it. Rooms such as kitchens and toilets should be maintained with the air at pressures less than the other spaces to promote air inflow into the room rather than outflow, which is likely to carry odors to other spaces.

343. Recirculation of Ventilating Air. During either winter or summer operation where it is necessary either to heat or cool air, a considerable saving in operating expenses can be secured by recirculating part of the air supplied to the spaces. In garage workshops and vehicular-tunnel ventilation it is not desirable to recirculate any of the air, as the primary purpose of ventilation is to dilute a deadly gas, carbon monoxide, CO, to a concentration which is not harmful to the occupants.

Recirculation of part of the ventilating air in winter requires less installed boiler and air-heater capacity and thereby not only reduces the first cost of the installation but also requires a smaller quantity of fuel, representing a considerable saving in operating expenses. Likewise in summer cooling the installed capacity of the air coolers can be made smaller and also the plant operating expenses reduced if a part of the air is recirculated.

The amount of air which may be recirculated depends upon the methods of treating it before it enters the rooms and also upon conditions within the rooms. If odors are a bothersome feature, more fresh air from the outside is necessary. Heat and moisture gains in the rooms may also require a greater amount of fresh air from the outside than would otherwise be required.

Generally speaking, it is desirable to reduce the amount of outside air drawn in to a minimum if the fresh air must be heated during the winter and cooled during summer operation. During mild weather, for either summer or winter operation, recirculation of the air may be omitted as the expenses of operation will not be excessive. The proportions of fresh and recirculated air vary somewhat but range from 20 to 30 per cent fresh air and 70 to 80 per cent recirculated air. Consideration should be give to the data of Table 103 when proportions of fresh and recirculated air are to be determined.

CHAPTER 15

HEATING WITH CENTRAL FAN-COIL SYSTEMS AND UNIT HEATERS

344. Fan-Coil Heating. For many years the scheme of heating which requires a fan, heater units, and a duct system was termed hot blast. This form of forced-convection indirect heating differs, for the most part, from mechanical warm-air furnace systems in equipment used for warming the air. Hot-blast heaters which have been used are assemblages of cast-iron sections, pipe coils, and copper and brass tubes with fins to give extended surfaces. Heaters built of cast-iron sections and sold under the trade name of "Vento" are in use, but their production has been discontinued. Likewise steel and wrought-iron pipe coils are not used in new layouts. Except in emergencies steel is not a common material for the tubes and fins of modern heating coils. The heat-conveying medium used within the heater sections is either steam or forced-circulation hot water, although steam is perhaps the more common of the two.

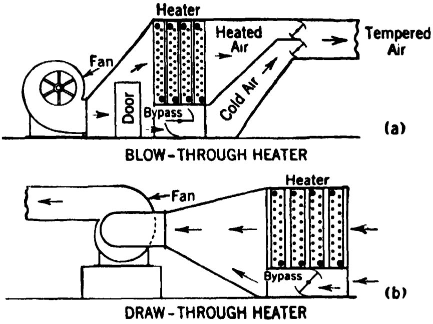


FIG. 282. Hot-blast heater arrangements.

The location of the heater, Fig. 282, with respect to the fan is either on its discharge side, a *blow-through arrangement*, or on its suction side which is termed a *draw-through* placement. The blow-through type of installation gives better facilities for by-passing part of the air around the heater, but it does not give the compact set-up that draw-through construction permits. With either heater arrangement the duct system attached may be either individual or trunk.

Central fan-coil apparatus not only serves for building heating alone, but it is also an integral part of central ventilating and air-conditioning systems. In some central ventilating and air-conditioning systems the heating of the air handled may be only sufficient to bring it to room temperature or thereabouts. Other central plants, which serve both as heating and as ventilating and air-conditioning systems, require that the heaters raise the air temperature to a value above that of the rooms heated.

As compared with heating systems using direct radiation, fan-coil apparatus has the following points in its favor: absence of direct radiators which occupy room space, requires a smaller amount of heating surface than direct radiation, ability to provide ventilation, and ease of air-temperature control.

345. Non-Ferrous Extended-Surface Blast Coils. Heater units consisting of either copper or brass tubes assembled between headers and having finned surfaces, mounted upon the tubes by either soldering or pressure contact or both, have the advantages of light weight, small space requirements, and ease of assembly. The production of the finished finned tube varies with different manufacturers. One coil is on the market in which the tubes have crimped spirally wound fins in firm mechanical contact with them, and the bonding is further made more effective by the application of solder to the parts to make the joint less resistant to the transfer of heat. Such tubes are fabricated with different outside diameters and with 3 and also 7 fins per linear inch of tube. Other makers of such equipment have different numbers of fins per unit of length and methods of fastening them to tubes.

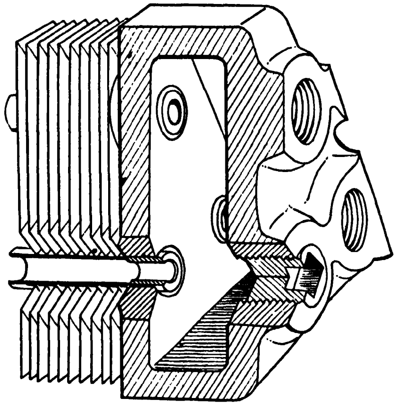


FIG. 283. Header arrangement of Trane blast coils.

A representative form of extended-surface heating-coil construction is indicated by the details of Figs. 283 and 284. These sections have semi-steel headers into which either copper or brass tubes are expanded by the use of internal bushings as shown by Fig. 283. The sections are built with one and two rows of tubes, as shown by Fig. 284; when two vertical rows of tubes are used the second row is staggered with respect to the first one. The fins are mechanically held against the tubes by a special expanding process, and they are neither brazed nor soldered to the tube; the mountings of the fins are as shown by Fig. 283. The number of fins per linear inch of tube varies with different sections. Thus for Trane Type-E sections, Series 66, 84, and

96 the number of fins per foot of tube length are 66, 84, and 96 which amounts to 5.5, 7, and 8 fins respectively per linear inch of tube length.

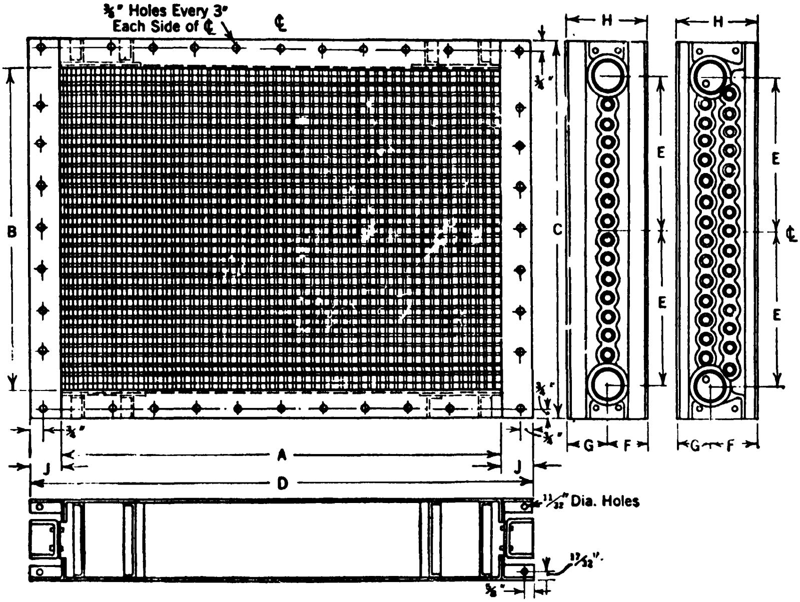


Fig. 284. Dimensions of Trane Type-E Series 96 blast coils.

The proper assemblage of a sufficient number of sections gives the required face area which should be as near a square as possible. In order to secure this shape of heater opening it may be necessary to arrange the sections in tiers as shown by the piping diagram of Fig. 285. The number of rows of tubes to give the final required air temperature is secured by banking the sections as shown by Fig. 285. Face area is the gross cross-sectional area inside the casing. The section casings are flanged and drilled, Fig. 284; when bolted together they do not require any other housing and are easily attached to the duct work.

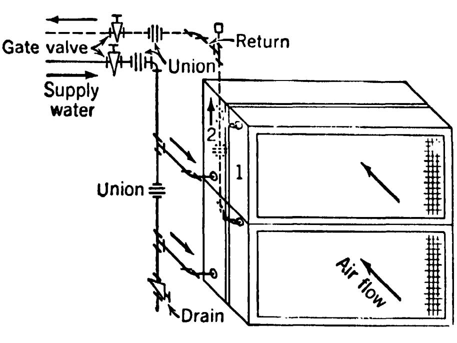


Fig. 285. Extended-surface heaters arranged in two tiers and two banks for hot-water counterflow operation.

TABLE 105

PHYSICAL DATA FOR TRANE TYPE-E BLAST COILS, DUCTWORK INSTALLATIONS

Approximate Roughing in Dimensions

(All dimensions in inches; dimension A = nominal or ordering length)

A In.	12	18	24	30	36	42	48	54	60	66	72	78	84	90	96	102	108	114	120
D In.	Overall Dimension $D = A + 4\frac{5}{8}$ Except 30-in. Headers Where $D = A + 5\frac{5}{8}$ and 33-In. Header Where $D = A + 6\frac{1}{8}$																		

Nominal or Ordering Width	6-In. Header		9-In. Header		12-In. Header		15-In. Header		18-In. Header		21-In. Header		24-In. Header		30-In. Header		33-In. Header	
Rows of Tubes	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row	1 Row	2 Row
B	6	6	9	9	12	12	15	15	18	18	21	21	24	24	30	30	33	33
C	9	9	12	12	$16\frac{1}{2}$	$16\frac{1}{2}$	$19\frac{1}{2}$	$19\frac{1}{2}$	$22\frac{1}{2}$	$22\frac{1}{2}$	$25\frac{1}{2}$	$25\frac{1}{2}$	$28\frac{1}{2}$	$28\frac{1}{2}$	$34\frac{1}{2}$	$34\frac{1}{2}$	$37\frac{1}{2}$	$37\frac{1}{2}$
E	$2\frac{3}{8}$	$2\frac{3}{8}$	$3\frac{7}{8}$	$3\frac{7}{8}$	$5\frac{1}{2}$	$5\frac{1}{2}$	7	7	$8\frac{1}{2}$	$8\frac{1}{2}$	10	10	$11\frac{3}{4}$	$11\frac{3}{4}$	$13\frac{1}{2}$	$13\frac{1}{2}$	15	$15\frac{1}{8}$
F	3	$3\frac{1}{2}$	3	$3\frac{1}{2}$	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{5}{8}$	3	$3\frac{1}{2}$	3	$4\frac{7}{16}$	3	$4\frac{13}{16}$
G	3	$2\frac{1}{2}$	3	$2\frac{1}{2}$	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{3}{8}$	3	$2\frac{1}{2}$	3	$2\frac{13}{16}$	3	$2\frac{7}{16}$
H	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	$7\frac{1}{4}$	6	$7\frac{1}{4}$
J	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{5}{16}$	$2\frac{13}{16}$	$2\frac{13}{16}$	$3\frac{1}{16}$	$3\frac{1}{16}$
Number of Tubes	4	7	6	11	8	15	10	19	12	23	14	27	16	31	20	39	22	43
Size of Pipe Tap	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	2	2	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	3

Face Area, Sq Ft for Various Length and Widths

Nominal Coil Length, *A* In.	12	18	24	30	36	42	48	54	60	66	72	78	84	90	96	102	108	114	120
Nominal Coil Width *B*, In.	Net Face Area, Sq Ft. Net Free Area = Approximately 60 Per Cent of Face Area																		
6	0.5	0.75	1.0	1.25	1.5	1.75	2.0	2.25	2.5	2.75	3.0								
9	0.75	1.13	1.5	1.88	2.25	2.63	3.0	3.38	3.75	4.13	4.5								
12	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
15	...	1.88	2.5	3.13	3.75	4.38	5.0	5.63	6.25	6.88	7.5	8.13	8.75	9.38	10.0	10.63	11.25	11.88	12.5
18	...	2.25	3.0	3.75	4.5	5.25	6.0	6.75	7.5	8.25	9.0	9.75	10.5	11.25	12.0	12.75	13.5	14.25	15.0
21	...	...	3.5	4.38	5.25	6.13	7.0	7.88	8.75	9.63	10.5	11.38	12.25	13.13	14.0	14.88	15.75	16.63	17.5
24	...	...	4.0	5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	19.0	20.0
30	...	...	...	6.25	7.5	8.75	10.0	11.25	12.5	13.75	15.0	16.25	17.5	18.75	20.0	21.25	22.5	23.75	25.0
33	...	...	...	...	8.25	9.63	11.0	12.38	13.75	15.13	16.5	17.88	19.25	20.63	22.0	23.38	24.75	26.13	27.5

Heating Surface per Square Foot of Net Face Area: Series 96, 1 Row, 23.0 sq. ft; 2 Row, 49.9 sq ft.

TABLE 106

PERFORMANCE DATA FOR TRANE TYPE-E, SERIES 96, BLAST COILS

Steam pressure 5 psig

Steam temperature 227 F
Latent heat = 960 Btu

FV = face velocity of air, fpm, corrected to 70 *F**

All temperatures given in degrees Fahrenheit

Condensate given in pounds of water per hour per square foot of face area

Inlet Air Temperature	Rows of Tubes Deep	200 *FV*		300 *FV*		400 *FV*		500 *FV*		600 *FV*		700 *FV*		800 *FV*		1000 *FV*		1200 *FV*	
		Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate	Final Temperature	Pounds Condensate
−40	1	65	23.8	50	30.5	40	36.3	33	41.4	29	46.6	24	50.8	21	55.1	16	63.1	12	70.3
	2	138	40.4	112	51.6	96	61.5	84	70.1	77	79.0	69	86.2	63	93.4	55	107.0	48	119.2
	3	173	48.3	151	64.7	135	79.3	123	92.3	115	105.3	107	116.2	101	127.2	91	147.7	82	166.4
	4	197	53.7	177	73.6	162	91.6	150	107.5	142	123.3	134	137.1	126	150.3	115	176.0	106	199.0
	6	217	58.2	205	83.1	196	106.6	186	128.0	180	149.2	173	168.5	167	187.8	157	223.0	149	256.0
	8	224	59.8	217	87.2	212	114.0	204	138.0	200	162.3	194	185.0	188	207.0	180	249.2	173	289.5
−30	1	72	23.0	56	29.3	47	35.0	40	39.7	36	45.0	32	49.1	29	53.2	24	61.0	20	67.7
	2	142	38.9	117	49.7	101	59.2	89	67.2	82	76.2	75	83.1	70	90.1	61	103	55	115.0
	3	176	46.5	154	62.3	139	76.4	127	88.5	120	101.6	112	112.1	105	122.7	96	142.4	87	159.7
	4	198	51.7	179	70.8	165	88.3	152	102.8	145	119.0	137	132.2	130	145.2	120	169.4	111	191.8
	6	218	56.0	206	80.1	197	102.7	187	122.6	181	143.2	175	162.7	169	180.6	160	214.8	152	245.7
	8	224	57.6	217	83.8	213	110.0	205	132.2	201	156.4	195	178.3	191	200.0	182	240.0	175	279.0
−20	1	77	22.0	63	28.2	54	33.6	48	38.2	44	43.2	40	47.1	36	51.0	32	58.6	28	65.2
	2	145	37.3	121	47.8	106	56.9	95	64.8	88	73.1	81	79.8	76	86.5	68	99.2	61	110.5
	3	177	44.6	157	59.9	142	73.4	131	85.3	124	97.4	116	107.7	110	117.8	101	137.0	93	154.1
	4	199	49.6	181	68.2	167	84.8	156	99.4	149	114.2	141	127.0	134	139.3	124	163.1	116	184.2
	6	218	63.7	207	77.0	198	98.6	189	118.2	183	138.0	177	156.0	171	173.5	162	206.5	155	237.3
	8	224	55.2	218	80.6	213	105.2	206	127.5	202	150.3	196	171.1	192	191.8	184	231.0	177	268.0
−10	1	83	21.1	70	27.1	61	32.2	55	36.7	51	41.4	47	45.2	44	49.0	40	56.0	36	62.5
	2	148	35.8	125	45.8	111	54.6	100	62.2	94	70.2	87	76.6	82	83.0	74	95.0	68	106.0
	3	179	42.8	159	57.4	145	70.4	135	81.9	128	93.5	121	103.3	115	113.0	106	131.1	99	147.8
	4	200	47.6	183	65.4	170	81.5	159	95.5	152	109.8	144	122.0	138	133.7	128	156.0	120	176.7
	6	218	51.6	208	73.7	199	94.6	190	113.0	185	133.3	179	150.0	174	166.3	165	197.9	157	227.2
	8	224	53.0	218	77.4	214	101.2	207	122.7	203	144.4	198	164.3	193	183.8	186	221.0	179	257.0
0	1	89	20.2	77	25.9	68	30.8	62	35.2	58	39.6	55	43.3	52	47.0	48	53.8	44	59.7
	2	151	34.3	129	43.8	116	52.2	106	59.6	99	67.2	93	73.4	88	79.5	81	91.0	75	101.5
	3	181	41.0	162	54.9	149	67.3	139	78.4	132	89.5	125	99.0	120	108.3	111	125.7	104	141.5
	4	202	45.6	185	62.6	172	78.0	162	91.5	155	105.0	148	116.8	142	128.2	132	149.6	125	169.2
	6	218	49.4	208	70.5	200	90.5	192	108.8	187	126.8	181	143.5	176	159.7	167	189.5	160	218.0
	8	224	50.8	219	74.2	214	97.0	208	117.5	204	138.0	199	157.2	194	176.6	188	212.0	181	246.0

10	1	95	19.3	83	24.7	75	29.5	70	33.6	65	37.9	62	[illegible]	60	44.8	56	51.4	52	57.2
	2	154	32.7	134	41.9	120	49.9	111	56.8	105	64.2	99	70.0	94	75.9	87	87.0	81	97.0
	3	183	39.1	165	52.5	152	64.4	142	74.8	136	85.5	129	94.4	124	103.4	116	120.1	110	135.4
	4	203	43.5	187	59.8	174	74.4	164	87.1	158	100.3	151	111.3	145	122.2	136	143.0	129	161.8
	6	218	47.1	209	67.4	201	86.5	193	103.6	189	121.0	183	136.8	178	152.2	170	181.2	164	208.5
	8	224	48.4	219	70.8	214	92.4	208	111.9	205	132.0	200	150.0	196	168.0	190	202.5	183	235.0
20	1	102	18.5	90	23.7	82	28.2	77	32.1	73	36.2	70	39.5	61	42.9	64	49.2	60	54.7
	2	158	31.3	138	40.1	125	47.7	116	54.4	110	61.4	105	67.0	100	72.6	94	83.3	88	92.7
	3	185	37.4	168	50.3	156	61.5	147	71.6	141	81.8	134	90.3	129	98.9	122	115.0	115	129.4
	4	204	41.6	189	57.2	177	71.4	167	83.4	161	96.0	155	106.5	149	117.2	141	136.9	134	154.8
	6	219	45.0	210	64.6	202	82.6	195	99.4	191	116.0	185	131.0	181	145.6	173	173.5	167	199.2
	8	225	46.4	220	67.7	215	88.6	209	107.0	206	126.2	201	143.5	198	161.5	192	193.9	186	225.0
30	1	108	17.6	96	22.5	89	26.8	84	30.5	81	34.5	78	37.6	75	40.7	71	46.7	68	52.0
	2	162	29.8	143	38.2	130	45.4	121	51.7	116	58.4	110	63.7	106	69.0	100	79.0	95	88.1
	3	187	35.6	171	47.9	160	58.6	150	68.0	145	77.8	139	85.9	134	93.9	127	109.1	121	123.0
	4	205	39.6	191	54.5	179	67.6	170	79.4	164	91.2	158	101.5	153	111.1	145	129.8	138	147.0
	6	219	42.9	211	61.5	204	78.8	197	94.2	192	110.2	187	124.3	183	138.2	175	164.8	169	189.3
	8	225	44.1	220	64.5	215	84.0	210	102.0	207	120.0	203	136.8	199	153.0	193	184.0	188	214.0
40	1	114	16.7	103	21.4	96	25.4	91	29.0	89	32.8	85	35.7	83	38.7	79	44.4	76	49.4
	2	165	28.3	147	36.2	135	43.1	127	49.2	122	55.5	116	60.5	113	65.5	106	75.1	102	83.7
	3	189	33.8	174	45.4	163	55.6	155	64.7	149	74.0	143	81.6	139	89.2	132	103.7	126	116.8
	4	206	37.6	192	51.6	182	64.3	174	75.5	168	86.6	162	96.3	157	105.5	149	123.6	143	139.8
	6	220	40.7	212	58.3	205	74.7	199	89.7	194	104.8	189	118.2	185	131.3	178	156.2	172	179.5
	8	225	41.8	220	61.2	216	80.0	211	97.0	208	114.0	204	129.8	200	145.0	195	175.0	190	203.0
50	1	119	15.7	110	20.2	103	24.0	99	27.4	96	30.9	93	33.8	91	36.0	87	42.0	84	46.7
	2	168	26.7	151	34.2	140	40.7	132	46.5	127	52.4	122	57.2	119	62.0	113	71.2	108	79.2
	3	191	31.8	177	42.9	166	52.5	158	61.2	153	69.8	148	77.2	143	84.5	137	98.3	131	110.6
	4	207	35.5	194	48.9	184	60.6	176	71.4	171	82.0	165	91.0	160	100.0	157	117.0	147	132.0
	6	220	38.5	213	55.1	206	70.5	200	85.0	196	99.0	191	112.0	187	124.4	181	148.5	175	170.0
	8	225	39.6	221	57.9	216	75.3	212	91.7	209	107.8	205	122.5	201	137.5	197	165.8	191	192.0
60	1	126	14.9	116	19.1	110	22.7	106	25.9	103	29.2	100	31.9	9[illegible]	34.4	95	39.5	92	44.0
	2	171	25.2	155	32.3	145	38.4	138	43.8	133	49.4	128	54.0	125	58.4	119	67.0	115	74.6
	3	193	30.1	180	40.5	169	49.6	162	57.6	157	65.8	152	72.9	148	79.6	142	92.4	136	104.0
	4	208	33.5	196	46.2	186	57.2	179	67.2	174	77.4	169	86.0	164	94.1	157	110.1	152	124.5
	6	220	36.2	213	52.0	207	66.6	201	79.9	198	93.4	193	105.5	189	117.3	183	139.3	178	160.0
	8	225	37.3	221	54.8	217	71.0	213	86.4	210	101.8	206	115.8	203	129.4	199	156.0	193	181.0
70	1	132	14.0	123	17.9	117	21.3	113	24.3	110	27.4	108	29.9	106	32.5	103	37.2	100	41.3
	2	175	23.7	160	30.3	150	36.1	143	41.2	138	46.4	134	50.7	131	55.0	126	63.0	122	70.0
	3	195	28.3	182	38.0	173	46.6	166	54.2	161	61.8	157	68.4	153	74.9	147	87.0	142	97.7
	4	209	31.5	198	43.3	189	53.8	182	63.3	177	72.5	172	80.8	168	88.6	162	103.6	156	116.7
	6	221	34.1	214	48.8	208	62.6	203	75.1	199	87.6	195	99.1	192	110.2	186	131.3	181	150.5
	8	225	35.1	221	51.3	218	66.9	214	81.4	211	95.3	208	108.9	204	121.9	200	146.5	195	170.0

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* Free area velocity = face velocity × 1.67. Recommended maximum face velocities: public buildings, heating and ventilating—800 fpm; industrial applications—1200 fpm.

Data for the physical dimensions of Type-E Trane blast coils are given by Fig. 284 and Table 105. Information relative to the performance of a Series 96, Type-E Trane blast coil is given by Table 106 for an operating steam pressure of 5 psig and various velocities of air flow and inlet temperatures. The air-friction pressure losses for the same equipment are given by Table 107, and the data are for the conditions of air at 70 F and 29.92 in. of mercury pressure.

TABLE 107

FRICTION-PRESSURE LOSSES IN TRANE TYPE-E, SERIES 96, BLAST HEATING COILS, INCHES OF WATER

Air temperature 70 F, barometric pressure 29.92 in. of mercury, 96 fins per ft

Rows of Tubes	Face Velocity, Fpm							
	300	400	500	600	700	800	1000	1200
1	0.019	0.031	0.045	0.060	0.078	0.098	0.142	0.193
2	0.043	0.068	0.100	0.135	0.175	0.217	0.320	0.425
3	0.062	0.099	0.145	0.195	0.253	0.315	0.462	0.618
4	0.086	0.136	0.200	0.270	0.350	0.434	0.640	0.850
6	0.129	0.204	0.300	0.405	0.525	0.651	0.960	1.275
8	0.172	0.272	0.400	0.540	0.700	0.868	1.280	1.700
10	0.215	0.340	0.500	0.675	0.875	1.085	1.600	2.125
12	0.258	0.408	0.600	0.810	1.050	1.302	1.920	2.550
14	0.301	0.476	0.700	0.945	1.225	1.519	2.240	2.975

346. Required Face Area and Rows of Tubes of a Blast Heater. The selection of fan-coil or blast heaters is made on the required amount of face area and the depth of the heater expressed in rows of tubes. Face-area requirements are determined by the quantity of air to be handled per minute and the allowable face area velocity. Both the quantity of air handled and its allowable velocity are expressed in terms of standard air which is measured at a temperature of 70 F and a barometric pressure of 29.92 in. of mercury. Satisfactory face-area velocities are 600 fpm for public buildings and 1200 fpm in industrial plants. When the quantity of air to be handled, measured under standard conditions, is Q_s cfm and the allowable velocity, under standard conditions, is V_s fpm the required face area, in square feet, is $A_f = Q_s/V_s$.

The number of rows of tubes required is dependent upon the initial-air temperature, the final desired air temperature, the average temperature of the heating medium within the coil sections, and the permissible face-area velocity.

The performance of blast-heater coils is influenced by the laws of heat transfer as affected by air velocities; heating-fluid velocities;

logarithmic mean-temperature difference between the two media; conditions of their flow, that is, laminar or turbulent; and certain coefficients peculiar to the heater design. Where reliable data, based on performance tests, are available their use is much simpler than attempts made to calculate the heater performance based on the known laws of heat transfer. The data of Table 106 are based on operation with steam at 5 psig and various face-area velocities. When the steam pressures are more or less, the numerical values of the table covering final air temperatures and condensation rates per hour are changed for the different air velocities and initial-air temperatures.

When blast-heater sections are operated with forced-circulation hot water their thermal outputs are different than when operated at the same entering air temperatures and face-area velocities with steam at any given pressure. Some coils when used with hot water are fitted with mechanical devices to create a more turbulent condition of water flow with a resultant larger output of heat than would be

TABLE 108

HEAT-TRANSFER CAPACITIES TRANE TYPE-E, SERIES 96, BLAST COILS WITH HOT-WATER OPERATION

All capacities are expressed in Btu per hour, per degree of mean-temperature difference between the air and water, per row of tubes, per square foot of face area

Water Velocity Fps	Without Turbulators								With Turbulators							
	Air Velocity, Fpm								Air Velocity, Fpm							
	300	400	500	600	700	800	900	1000	300	400	500	600	700	800	900	1000
0.5	82	90	95	100	104	107	110	112	113	128	139	149	157	164	171	176
1.0	101	112	122	130	135	140	145	149	124	142	156	169	180	189	197	204
1.5	111	126	137	147	155	161	168	173	130	150	166	180	192	203	213	221
2.0	118	135	148	160	169	177	184	190	134	155	173	188	201	212	223	233
2.5	124	142	156	170	180	189	197	205	137	159	178	194	207	220	232	242
3.0	128	147	163	178	188	198	207	215	139	162	181	198	213	226	238	250
4.0	134	155	173	190	201	213	224	233	143	167	188	206	222	236	250	261
5.0	138	161	180	198	211	224	236	247								
6.0	142	166	186	205	219	233	246	257								

the case with the normal conditions within the coils. The data of Table 108 are expressed in Btu per hour, per row of tubes, per degree of mean-temperature difference, per square foot of face area and are given for sections with and without turbulators.

As an aid in the rapid determination of water velocities in Trane Type-E coils with two rows of tubes the data of Fig. 286 are given. When coils having only one row of tubes are used the velocities read from the graphs of Fig. 286 are multiplied by a factor of 1.9.

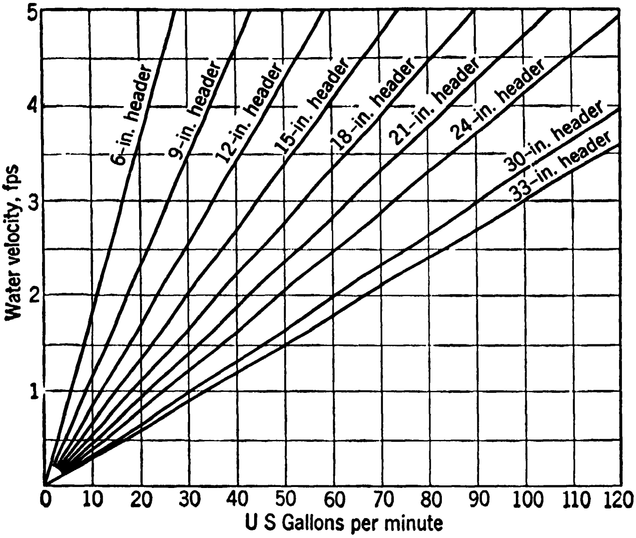


FIG. 286. Water velocities in Trane Type-E coils with two rows of tubes. For one-row coils, multiply above water velocities by 1.9.

When a blast coil is used for air heating with forced-circulation hot water as a heating medium the resistance which the coil offers to the water flow is of importance. The water friction losses, in feet head of water flowing, are given for Trane Type-E coils of different section lengths and velocities of flow by Table 109.

For finned-tube non-ferrous heaters the condensation per hour is equal to the square feet of face area times the condensation per square foot of face area. The condensation rate for heaters is obtained from tables of performance. The total condensation per hour can also be determined on the basis of the weight of air, its temperature rise, and the latent heat, or enthalpy of evaporation, of the steam used, as shown by equation 137.

$$W_s = \frac{0.24W_a(t_2 - t_1)}{h_{fg}} \tag{137}$$

TABLE 109

Friction Losses, Feet Head of Water, Trane Type-E Blast Coils When Operated with Water*

Water Velocity, Fps	Without Turbulators										With Turbulators									
	Coil Face Length, In.										Coil Face Length, In.									
	12	24	36	48	60	72	84	96	108	120	12	24	36	48	60	72	84	96	108	120
0.5	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.04	0.05	0.07	0.09	0.10	0.12	0.14	0.15	0.17	0.19
1.0	0.09	0.11	0.12	0.13	0.15	0.16	0.18	0.19	0.21	0.22	0.13	0.18	0.23	0.28	0.33	0.38	0.43	0.48	0.53	0.58
1.5	0.19	0.22	0.25	0.28	0.31	0.34	0.37	0.40	0.43	0.46	0.25	0.35	0.45	0.55	0.65	0.75	0.85	0.95	1.10	1.20
2.0	0.32	0.37	0.42	0.47	0.53	0.58	0.63	0.68	0.73	0.79	0.42	0.58	0.74	0.90	1.10	1.20	1.40	1.50	1.70	1.90
2.5	0.48	0.56	0.64	0.71	0.79	0.87	0.95	1.00	1.10	1.20	0.63	0.85	1.10	1.30	1.60	1.80	2.00	2.20	2.40	2.70
3.0	0.67	0.78	0.89	1.00	1.10	1.20	1.30	1.40	1.60	1.70	0.86	1.20	1.50	1.80	2.10	2.40	2.70	3.00	3.30	3.60
3.5	0.89	1.00	1.20	1.30	1.50	1.60	1.80	1.90	2.10	2.20	1.10	1.50	1.90	2.30	2.70	3.10	3.50	3.90	4.30	4.70
4.0	1.20	1.30	1.50	1.70	1.90	2.10	2.30	2.50	2.60	2.80	1.40	1.90	2.40	2.90	3.40	3.90	4.40	4.80	5.30	5.80
5.0	1.70	2.00	2.30	2.60	2.80	3.10	3.40	3.60	3.90	4.20										
6.0	2.40	2.80	3.20	3.60	4.00	4.40	4.70	5.10	5.50	5.90										

---

* When two or more coil sections are used 30 per cent should be added for friction in intermediate piping connections.

where W_s = weight of steam per hour, lb.
W_a = weight of air handled per hour, lb.
t_1 = temperature of air entering heater, deg F.
t_2 = temperature of air leaving the heater, deg F.
h_{fg} = enthalpy of evaporation of steam used in the heater, Btu per lb.

347. Correction of Heater Air Friction-Pressure Losses to Actual Conditions. The air friction losses through the heater at standard conditions must be corrected to actual conditions. In order to do this the mean temperature of the air as it passes through the heater is required in making the correction, as the average air density is necessary. This mean temperature can be obtained from the equation

$$t_s - t_m = \frac{t_2 - t_1}{\log_e\left(\dfrac{t_s - t_1}{t_s - t_2}\right)} \tag{138}$$

where t_s = temperature of the heating medium in the heater, deg F.
t_m = mean temperature of the air as it passes through the heater, deg F.
t_1 = temperature of the air entering the heater, deg F.
t_2 = temperature of the air leaving the heater, deg F.
$t_s - t_m$ = mean temperature difference between the heating medium and the air, deg F.

$$t_m = t_s - \frac{(t_2 - t_1)}{\log_e\left(\dfrac{t_s - t_1}{t_s - t_2}\right)} \tag{139}$$

When the mean temperature of the air, t_m, has been computed, the mean density of the air, d_m, is easily obtained. Examination of equation 132, for friction losses, h_f, reveals that certain portions of the equation may be regarded as practically constant for air at 70 F or at any other mean temperature. These portions of the equation are $f \times (LR/A) \times (12/2gd_w)$. The coefficient of friction for the heater does change slightly with change of air velocity because of temperature increase, but the change is negligible here. The factors v and d are changed by a change of air temperature. For air at standard conditions where the air friction loss h_{fs} is known the equation may be stated as

$$h_{fs} = f \times \frac{LR}{A_f} \times \frac{12d_s}{d_w} \times \frac{v_s^2}{2g}$$

where A_f = heater free area, sq ft.
v_s = free area velocity of air at standard conditions, fps.
d_s = density of air at standard conditions, lb per cu ft.

The weight of air handled, expressed in terms of the free area of the heater, the air velocity, and the air density for both standard and actual conditions, is $W_a = A_f \times d_s \times v_s = A_f \times d_a \times v_a$ lb per sec, and $v_a = v_s \times \frac{d_s}{d_a}$. Hence equation 132 may be rewritten for actual conditions:

$$h_{fa} = \frac{fLR}{A_f} \times \frac{12d_a}{d_w} \times \frac{\left(v_s \frac{d_s}{d_a}\right)^2}{2g}$$

$$= \frac{fLR}{A_f} \times \frac{12d_s}{d_w} \times \frac{v_s^2}{2g} \times \frac{d_s}{d_a}$$

but the portion of the equation $(fLR/A_f) \times (12d_s/d_w) \times (v_s^2/2g)$ is obtainable from published data for standard conditions so that the corrected air friction-pressure loss through the heater is

$$h_{fa} = h_{fs} \times \frac{d_s}{d_a} = \text{in. of water} \qquad (140)$$

Equation 138 covers calculations where the temperature of the heating medium, steam, remains constant. When hot water is used as a transfer medium its temperature drops, as some of its heat is given to air passing through a blast heater. In order to maintain the maximum temperature difference between the water and the air and thus facilitate the transfer of heat in the coil, the counterflow method of operation is generally used. The principle involved is feeding the hottest water into the heater where the air leaves with its highest temperature and having the discharge water, at its lowest temperature, leave the heater at the air-inlet end where the air has the least amount of heat. For such conditions the mean temperature difference between the water and the air being handled is

$$t_{wm} - t_{am} = \frac{(t_{w2} - t_{a1}) - (t_{w1} - t_{a2})}{\log_e \frac{(t_{w2} - t_{a1})}{(t_{w1} - t_{a2})}} \qquad (141)$$

where t_{wm} = mean temperature of water, deg F.
t_{am} = mean temperature of air, deg F.
t_{w1} = temperature of entering water, deg F.

t_{w2} = temperature of leaving water, deg F.
t_{a1} = temperature of entering air, deg F.
t_{a2} = temperature of exit air, deg F.

When the mean temperature of the air, t_{am}, is desired it may be calculated by use of equation 141, with a small error, taking t_{wm} as the numerical mean of t_{w1} and t_{w2}.

348. Selection of a Blast Heater. The system used in this discussion will be that of Fig. 287 with the quantities of air as indicated and the following duct diameters: section one, 27 in.; two, 23.5 in.; three, 20 in.; and four, 14 in.

Example. A building is to be heated by a draw-through fan-coil system as indicated by Fig. 287. The air temperature desired in the occupied zone of the space is 60 F, and the air is to enter it at an average temperature of 130 F. One-fourth of the air handled is to be taken as fresh air from the outside at −20 F. The air will leave the heater at a temperature approximately 2 F above the average duct-air temperature. The building heat losses are 330,800 Btu per hr when the inside-air temperature is 60 F and the outside-air temperature is −20 F. The heater is to operate with a face-area velocity of 1000 fpm at standard conditions for the air. The actual friction losses in the longest run of the duct are 0.114 in. of water, the losses in the air filter 0.200 in. of water, and in the heater to fan-inlet transition piece 0.05 in. of water.

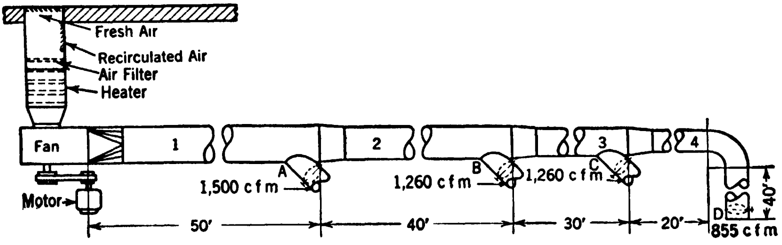


FIG. 287. Hot-blast heating system with air filters.

(*a*) Find the required size of the heater when it is operated with steam at 227 F and the static pressure against which the fan must operate. (*b*) Find the size of the heater when it is to function with water entering it at 210 F and leaving it at 185 F. Determine the static pressure which the fan must overcome.

Solution. (*a*) W_a, the weight of air handled per hour, is $330{,}800/0.24(130 - 60) = 19{,}690$ lb. The weight of ventilating air, W_v, is $19{,}690/4 = 4922$ lb per hr, and the weight of recirculated air thereby becomes $19{,}690 - 4922 = 14{,}768$ lb per hr. The average temperature of the air entering the heater is computed as $[(14{,}768 \times 60) + (4922 \times -20)]/19{,}690 = 40$ F. Dry air at a temperature of 70 F and a pressure of 29.92 in. of mercury has a density of 0.07492 lb per cu ft. The volume of the air entering the heater under standard conditions is $19{,}690/(60 \times 0.07492) = 4375$ cfm.

The required face area of the heater is $\frac{4375}{1000} = 4.38$ sq ft. Inspection of Table 105 indicates that a single section with a 21-in. header and a coil length of

30 in. provides this area. Although the air-inlet opening is not square its rectangular shape is near enough to that of a square so that the air as it is drawn through it by the fan will have a satisfactory distribution over the heating surfaces. Table 106 shows that for the conditions indicated three rows of tubes are required to give the final air temperature of 132 F. This heater depth may be secured by employing one 2-row and one 1-row tube heater section assembled together to give three rows of tubes over which air flow will occur. For these conditions of operation the condensation is given by Table 106 as 103.7 lb of condensate per sq ft of face area per hr. The weight of steam required per hour may be estimated as either $103.7 \times 4.38 = 454$ lb or $[0.24 \times 19{,}690(132 - 40)] \div 960.7 = 453$ lb.

The mean temperature of the air passing through the heater is $t_{am} = 227 - \{[(132 - 40)]/\log_e[(227 - 40) \div (227 - 132)]\} = 91.2$ F, and d_a at this temperature is 0.0721 lb per cu ft. Table 107 indicates that the friction-pressure losses resulting from air flow through the heater are 0.462 in of water under standard conditions, for this problem. The heater friction-pressure losses under actual conditions of air flow are $0.462 \times (0.07492 \div 0.0721) = 0.480$ in. of water.

The entrance losses at the fresh-air intake will be taken as 1.25 times the velocity pressure of the air passing through the inlet grille. The inlet will be designed so that the free-area velocity in the grille will not exceed 15 fps with air at −20 F. Equation 122 applied to this case expresses the fresh-air inlet pressure loss or resistance as $1.25(12 \times 0.0902 \times 15 \times 15) \div (62.3 \times 2 \times 32.16) = 0.076$ in. of water. The recirculating air inlet will be larger than the outdoor inlet, and the density of the fresh air will be greater than that recirculated. Therefore the dampers at the two openings must be adjusted to give the proper air quantities, and the inlet resistance to air flow will be the calculated 0.076 in. of water.

The static pressure against which the fan operates becomes, therefore, the sum of the frictional resistances to be overcome on both its suction and its discharge side. The static pressure h_s is $0.076 + 0.200 + 0.050 + 0.480 + 0.114 = 0.920$ in. of water. The necessary fan and the power required to drive it may be ascertained by use of Table 93, Chap. 13.

(*b*) When the unit is to operate with hot water the required face area is determined as for steam and the calculations of (*a*) apply. The procedure for determining the necessary rows of tubes necessitates the estimation of the load on the heater, which is $H = 0.24 \times 19{,}690(132 - 40) = 434{,}750$ Btu per hr. The mean-temperature difference between the water and the air as found by equation 141 is $t_{wm} - t_{am} = [(185 - 40) - (210 - 132)] \div \log_e[(185 - 40) \div (210 - 132)] = 108$ F. The U.S. gallon contains 231 cu in., or one cubic foot is equivalent to $1728 \div 231 = 7.48$ gal. The quantity of water required by the heater, as measured at 210 F, is $(434{,}750 \times 7.48) \div [(210 - 185)59.88 \times 60] = 36.1$ gpm. In Fig. 286 the velocity of flow is shown to be 1.7 fps for a Trane Type-E section with a 21-in. header. With the velocity existent the use of turbulators is desirable so that by interpolation of Table 108 data a heat-transfer factor of 226 is found for the conditions of velocities of 1.7 fps for water and 1000 fpm for air. The heat given off per hour per row of tubes is the product of the required face area, the mean-temperature difference of the water and the air, and the heat given off per hour per tube per square foot of face area. The number of tubes required is $434{,}750/(4.38 \times 108 \times 226) = 4.06$, which will be used as 4. The heater necessary can be made up of two 21 by 30 in. sections, each having two rows of tubes, so placed or banked that the air will flow over 4 rows of tubes. Using the mean-temperature difference of the water and air as 108 F, and 197.5 F as the average of the initial and the final water temperatures, the average air temperature may be taken as 89.5 F at which its density is 0.07219 lb per cu ft when dry. A 4-tube heater has air-friction losses of 0.640 in. of water under standard conditions for the velocity specification of the problem. The correction to actual conditions of operation is (0.640 × 0.07492)/0.07219 = 0.664 in. of water. With the exception of the air-friction losses in the heater the other resistances to air flow are the same as those of (*a*). The calculated static pressure which the fan must overcome is 0.076 + 0.200 + 0.050 + 0.664 + 0.114 = 1.104 in. of water. The data of Table 109 are such that interpolations of them may be made over limited ranges. On this basis the friction losses of water flowing through a 21 by 30 in. section, when the velocity is stated to be 1.7 fps, may be estimated as 0.50 ft per 2-row section fitted with turbulators. Four rows of tubes are involved in the use of the two-row sections. Losses for the sections should include an extra allowance of 30 per cent for intermediate piping. An estimation of head to be overcome by a pump, as offered by the coil sections and their connections, is 0.50 × 2 × 1.3 = 1.30 ft.

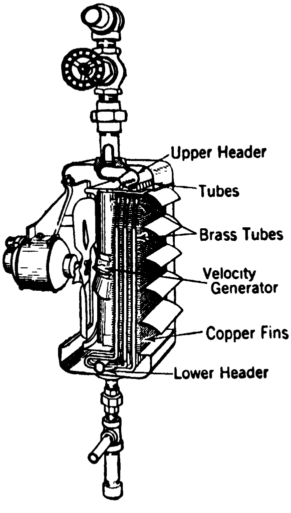
349. Unit Heaters. An assemblage of an air-heater section and a fan, which circulates air over it, within an enclosing housing embodies the essential requirements of a unit heater. Generally the design is so constituted that the complete unit, which is factory assembled, can be quickly installed in place with the minimum amount of work. For the most part no duct work or a very small amount of duct work is attached to unit heaters. When such a provision is made either the duct may be only a short connection through an outside wall for the purpose of bringing in ventilating air to be warmed by the heater, or it may be merely an attempt to draw cool air from a floor when the heater is placed some distance above it. This arrangement is an aid in securing good distribution of heated air within a space.

Classifications of unit heaters may be distinguished by (1) the heating medium, that is, steam, hot water, electricity or the products of combustion of gas, oil, or coal burned within them; (2) the type of fan used for air circulation, which may be either a propeller or a centrifugal unit; and (3) the arrangement of the heating surfaces with respect to fan location: either blow-through or draw-through arrangement.

Low- and high-pressure steam and mechanically circulated hot water commonly serve as heating fluids which may be utilized in the unit some distance from the boiler where they receive heat from a burning fuel. Many installations employ gas as a fuel that is burned within the unit where transfer is made, through the walls of flue-gas passageways, of heat to the air handled by the fan. The same is true of oil and coal-fired installations. The latter may be either hand or stoker fired. Electric motors are commonly used to operate heater fans. However, steam turbine drives are available with the exhaust steam discharged to and condensed within the heater coils.

Unit heaters have wide applications for the heating of industrial buildings, machine and other workshops, garages, stores, offices, lobbies, the heating of air for drying materials, and the heating of air to dissipate fogs which may arise from water vaporized in manufacturing processes. Compared with central-fan-coil heating systems and those equipped with direct radiators the advantages of unit heaters lie in the matter of both their first cost and the space required in the building. Other factors in the favor of unit heaters are their light weight, capacity ranging from small to large, ability to reduce air stratification by its circulation, rapidity of heating, and flexibility of air-temperature control together with reduced costs of heating and system simplification.

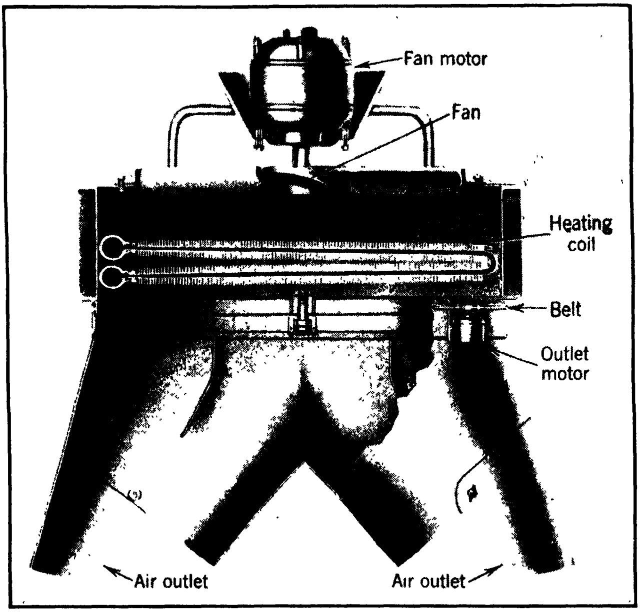
350. Types of Unit Heaters. The main classifications of unit heaters are the suspended and the floor types. Suspended heaters may have either propeller or centrifugal fans, and their position is generally the more common blow-through location rather than that of drawing the air through the heating element. The air flow of unit heaters may be in either a vertical or a horizontal direction. Vertical downward discharge units may have either stationary or revolving air outlets. The latter are provided to give better distribution of the air in the space beneath them.



(*Modine Manufacturing Co.*)
FIG. 288. Suspended-type unit heater.

351. Suspended Unit Heaters. The support of such a unit heater may be either from the steam or water-supply piping, Fig. 288, or otherwise as in Fig. 291; in any event the heater is placed above the zone of occupancy. This type of heater usually has some kind of lightweight finned tubing in the heating element as shown by Figs. 288 and 289. which illustrate units fitted with blow-through propeller fans. The discharge outlets of the vertical suspended unit heater of Fig. 289 are rotated by a combination of a belt and an electric motor. Figure 290 illustrates the essential parts of a gas-fired unit heater, which is a suspended type of unit with a propeller fan producing horizontal discharge of the air. Similar units are available for installation in short horizontal ducts.

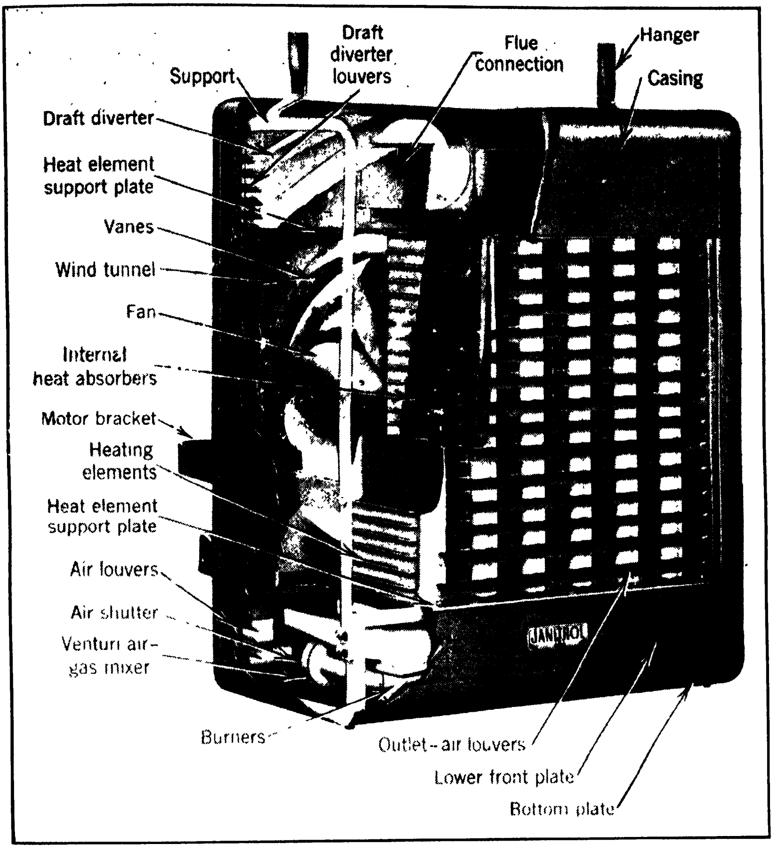
Suspended unit-heater installations are sometimes made with fresh-air connections from the outside together with provisions for the recirculation of room air as indicated by the arrangement of Fig. 291, which is fitted with a centrifugal fan. When the air dampers of the unit, Fig. 291, are properly adjusted and its heater coil inoperative the equipment can be used as a ventilator to withdraw air from a space and to discharge it outside the structure. The installation shown by Fig. 292 is that of a suspended unit heater with a propeller fan and, arranged to take air from near the floor level as well as from outdoors.



(*L. J. Wing Manufacturing Co.*)

FIG. 289. Vertical-discharge suspended unit heater with revolving outlets.

352. Floor Types of Unit Heaters. A typical floor-type unit heater is shown by Fig. 293. The unit consists of a housing with an air inlet at the floor and air outlets at the top which may be arranged to discharge in two directions. The centrifugal fans draw the air to be heated through the heating coils placed just above the air inlet opening near to the floor. The air discharged may leave its outlets at either moderate or high velocities just above the top level of the occupied zone. Figure 294 diagramatically shows the arrangement of a floor-type gas-fired unit heater with a centrifugal fan discharging the air to be heated through the heat-exchanger section of the unit. The stoker-fired coal-burning unit of Fig. 295 is a floor-type factory-assembled unit which may be readily moved from place to place by means of



(*Surface Combustion Corp.*)

FIG. 290. Gas-fired suspended unit heater.

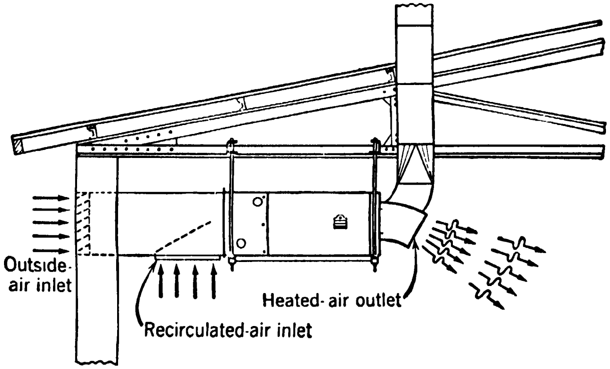


FIG. 291. Centrifugal-fan suspended unit heater arranged for ventilation and air recirculation.

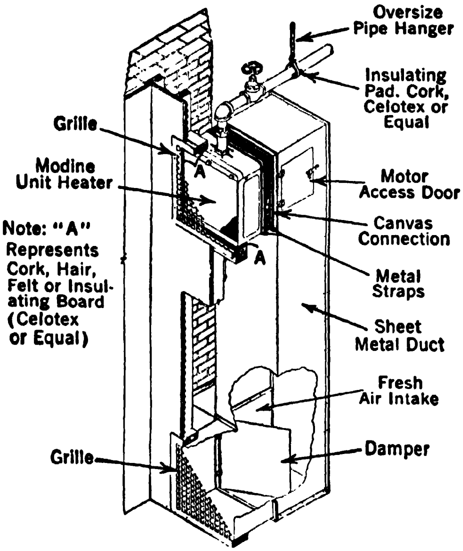
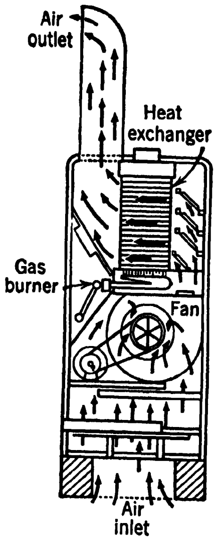
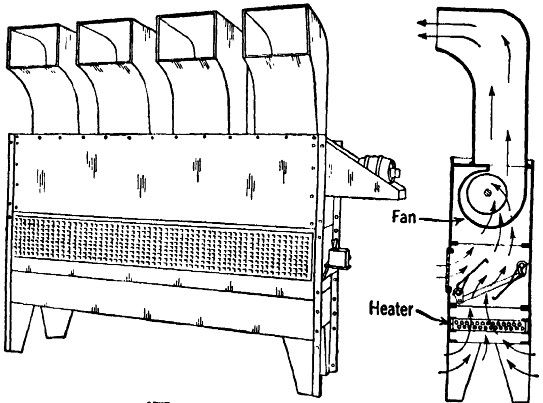


Fig. 292. Exterior unit-heater installation.



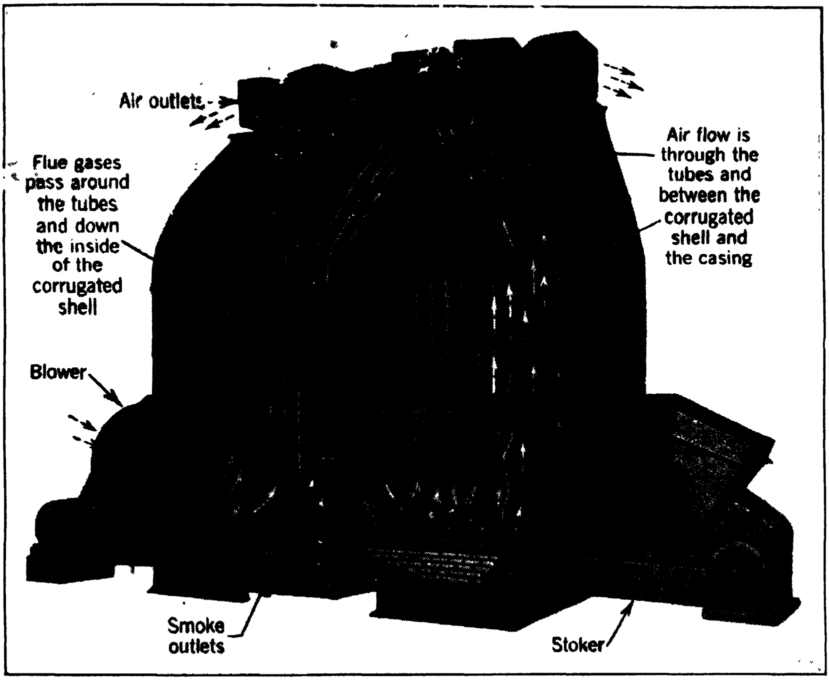
(*Surface Combustion Corp.*)
Fig. 294. Floor-type gas-fired unit heater.



(*Westinghouse Electric Corp., Sturtevant Division*)
Fig. 293. Floor-type unit heater.

a crane. These heaters do not require foundations. This stoker-fired unit heater may also be provided with a duct system and used as a mechanical warm-air furnace system if desired.

Fresh outside-air connections may be made to floor-type unit heaters. Floor units require space in the occupied zone; they give fairly uniform air temperatures in this location but care must be exercised that the space occupants are not subjected to disagreeable air motion in any spot.


Air outlets
Flue gases pass around the tubes and down the inside of the corrugated shell
Air flow is through the tubes and between the corrugated shell and the casing
Blower
Smoke outlets
Stoker

(*Lee Engineering Co.*)

FIG. 295. Tubular stoker-fired unit heater.

353. Air Discharge from Unit Heaters. The velocities of air leaving unit heaters are dependent upon their type and location. The velocities of air leaving units may range from 300 to 2500 fpm, and the warmed air may be made to travel from 75 to 250 ft. Outlet-air velocities applicable to unit heaters are as follows: centrifugal fans with horizontal discharge, 1500 to 2500 fpm; propeller-fan units with horizontal discharge, 400 to 1000 fpm; and vertical-discharge units, 1200 to 2200 fpm with corresponding distances of air flow in feet, respectively, equal to 20 to 250, 30 to 100, and up to 75. In addition to the direction of air into the occupied zone it is advisable to arrange unit heaters to give rotational circulation of the air in the space. This is accomplished by placing the outlets of all heaters to blow at an angle with the side walls of the building so that all heaters working together will produce a gentle movement of all the air in the space. Possible locations of unit heaters are given by Fig. 296. Any of the arrangements shown can be made to give satisfactory service. When all the heaters are placed in the middle of the space or along one wall the problem of the supply and return piping is simpler than where unit heaters are placed along all outside walls.

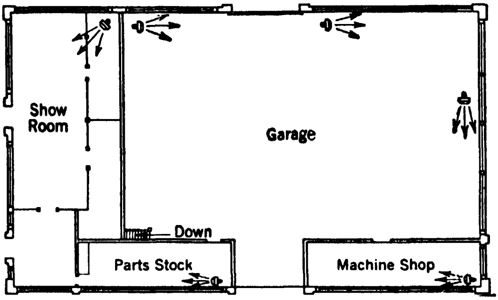
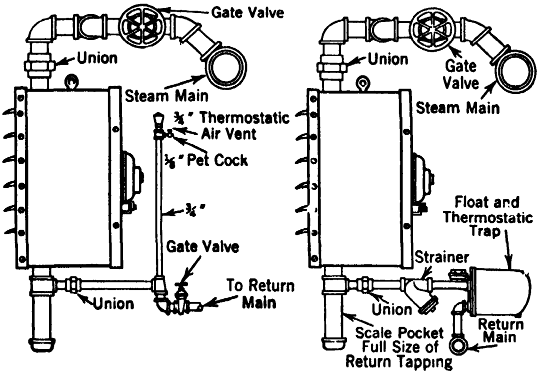


Fig. 296. Unit-heater placement.



(*C. A. Dunham Co.*)

Fig. 297. Piping details of unit heaters.

354. Supply-Piping Details. All steam and water-filled unit heaters require a separate supply and return connection to the heating element. Therefore the common use of the steam unit heaters is with a two-pipe system. When a one-pipe gravity-return steam system has unit heaters as part of its heat dispersal units, no attempt is made to have the condensate drain from the heater to the steam main. A drain connection from the heater is run to a wet-return main and an ample clearance maintained between the heater element and the water level as it stands in the drip piping. Unit heaters may also be used with one-pipe steam-heating systems where a condensation receiver and pump are installed as in a mechanical system. Traps are used at the heater outlets to seal off the steam from the drain piping, and the pump is used to return the condensate to the boiler.

Care must be taken in installing the piping for the heat carrier, steam or water, to choose large-enough pipes. The venting of air from the heaters and piping and the removal of water from the heater coils are very important. Representative examples of steam connections to unit heaters are detailed in Fig. 297.

355. Rating of Unit Heaters. Codes,[1] as formulated by the American Society of Heating and Ventilating Engineers and the Industrial Unit Heater Association, are of value in establishing ratings for steam and water filled units. The code[2] sponsored by American Gas Association applies to gas-fired unit heaters.

All unit heaters of the various types are rated on the basis of the heat delivered by the air emerging from them as measured in Btu per hour above an entering air temperature of 60 F. When attempts are made to compare the performance of one heater with another consideration must be given to other criteria which include the temperature of the heating element and the velocity of the air passing through it.

The basis for rating steam and water unit heaters is air entering them at 60 F and a barometric pressure of 29.92 in. of mercury when either dry saturated steam is used at 2 psig or heated water enters them at 200 F. In either case the heater is to operate free of external resistances to the flow of air through it. Oil-fired and gas-fired heaters are rated in terms of output in Btu per hour, and the gas-using equipment also has a further rating expressed in terms of Btu per hour input. Coal-fired units are rated on the basis of output, Btu per hour, as measured at the discharge outlets. Electric heaters are rated in terms of the hourly input to the unit in either kilowatts, Btu, or square feet of equivalent direct radiation.

356. Air-Temperature Control. Manual control of the fan motor of a unit heater may be all that is required either with a single-speed motor or with capacitor motors having two or more operating speeds. With variable-speed motors, hand control may give almost continuous operation of the fan, and the steam is not shut off when the fan is not running. The output is also controlled by the action of a room thermostat which starts and stops fan motors. Temperature is also regulated by means of a thermostatically controlled valve in the steam supply line to vary the amount of steam admitted to the heater coils. With this method of control a thermostat placed in the return line is desirable to prevent the operation of the fan motors when steam is not being supplied to the coils.

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[1] "Standard Code For Testing and Rating Steam Unit Heaters," 1930. "Standard Code For Testing Hot Water Unit Heaters," 1942.

[2] "American Standard Approval Requirements for Gas Unit Heaters," 1940.

357. Selection of Unit Heaters. As in all heating systems a careful estimate of the building hourly heat requirements is necessary. This estimate must include not only the heat losses from the spaces but also the heat to be added to ventilating air brought in from the outside.

The type of heater to choose depends upon the building construction as regards distances of blow permissible and the spaces available for heaters. In general, it is expedient to handle larger quantities of moderately heated air rather than smaller highly heated volumes, as the distribution problems are simpler with reasonably low temperatures. The number and sizes of the heaters used are dependent upon the local conditions of operation. When the air entering the heater is taken from near the floor level its temperature is considered to be that prevailing at the breathing-line level.

The spacing of unit heaters depends upon the outlet-air velocity and temperature. High-velocity heaters may be expected to give greater distances of air flow. High-velocity floor units may be spaced up to 250 ft apart and suspended-type heaters with air velocities of 250 to 500 fpm anywhere from 50 to 100 ft apart. With any type, the spacing should be such that with the outlet velocity existent the maximum amount of induced air circulation in the occupied space will be created by the air leaving the heaters.

PROBLEMS

1. A blast coil with two rows of tubes and listed as 33 by 36 in. is operated with steam at 227 F and air passing its face with a velocity of 1200 fpm under standard conditions. The barometric pressure is 28.5 in. of mercury, and the air enters the heater at 30 F. Under the conditions specified find (*a*) the weight of air that the heater will warm each hour, (*b*) the final air temperature to be expected, and (*c*) the weight of dry steam, in pounds per hour, by two methods.

2. A blast heater has three rows of tubes and is made up of 21 by 42 in. sections two tiers high. The conditions of heater operation are: barometric pressure, 29.0 in. of mercury; steam pressure, 5.4 psig; temperature of air entering heater, 10 F; and face velocity of air, 500 fpm standard conditions. Determine (*a*) the weight of air being handled each hour, (*b*) the final air temperature, and (*c*) the weight of steam used each hour as determined by two methods.

3. Find the actual air friction-pressure losses for the operating conditions of both problem 1 and 2.

4. A blast heater is to operate with water entering it at 200 F and leaving at 175 F when 15,760 lb of air are to be heated per hour from 50 to 143 F with a face-area velocity of 800 fpm, at standard conditions, when the barometric pressure is 29.3 in. of mercury. Select a heater and find its air-friction losses and water-pressure loss.

5. Determine the requirements of a blast heater which is to handle 20,000 cfm of entering air measured at 40 F when the barometric pressure is 29.5 in. of mercury if operated with a final air temperature of 135 F when the face velocity is 600 fpm at standard conditions. Water is to enter the heater at 205 F and leave it at 185 F. Determine the dimensions of the heater, the required number of rows of tubes, the water-friction losses in the heater and the air-friction losses through the heater sections.

6. Blast-heater coils are to be used in a blow-through unit in a factory building. The air of the space is to be maintained at 60 F, and the air is to leave the duct outlets at a temperature of 140 F. The heat losses from the building are 1,245,000 Btu per hr when the outside-air temperature is −20 F. A loss of temperature of 3 deg F will occur in the duct system, and the heat thus lost will not be available for heating the building. The allowable face-area velocity at standard conditions is 1200 ft per min. The barometric pressure is to be taken as 29.00 in. of mercury, and saturated steam is available for use in the coils at a temperature of 227 F. One-fourth of the air to be handled is to be from the outside, and the remainder is to be room air recirculated. Find (*a*) the number of heater sections and the number of rows of tubes required, (*b*) the face area of the heater, and (*c*) the weight of dry steam required per hour. Check the weight of steam by two methods.

7. The fan-blast system of Fig. 287 is to operate in a building having heat losses equal to 950,000 Btu per hr when the inside-air temperature is 65 F and the outside-air temperature is 0 F. The loss of air temperature in the ducts amounts to 3 deg F, and the heat losses from the ducts are not available for heating the building. One-third of the air passing through the heater is to be taken from the outside, and the remainder of the air handled is to be recirculated air. The quantities of air handled in the various sections of the duct system are to be directly proportional to the quantities specified by Fig. 287 and the average outlet-air temperature is to be 125 F. The barometric pressure is 29.92 in. of mercury, and steam for heating is available at a temperature of 227 F. (*a*) Design a duct system with outlet velocities of 900 fpm giving the sizes for the conditions of this problem. (*b*) Determine the required number of finned-heater sections necessary if the free area velocity is approximately 1200 fpm at standard conditions. (*c*) Find the static pressure against which the fan must work if the entrance losses at the fresh-air intake and the recirculated-air intake are 1.25 times the velocity pressure of the air as it passes through the free area of each inlet opening. Take the allowable free-area velocity is 1000 fpm under actual conditions of flow, and consider the free area as 50 per cent of the gross area of each inlet opening. (*d*) Select a fan for this installation. (*e*) Find the horsepower of the motor to drive the fan if it is belt driven and the efficiency of the drive is 98 per cent.

8. A kiln for drying materials must receive 10,800 cfm of air at 160 F when the barometric pressure is 29.40 in. of mercury. All the air handled is taken from the outside at 0 F. The suction duct of the fan is rectangular and has the same cross-sectional area as the face of the non-ferrous extended-surface heater which is to operate with a face-area velocity at 1200 fpm. The length of the inlet duct is 10 ft, and it has no elbows. The entrance losses to the inlet duct are 1.5 times the

velocity pressure of the air as it passes through the inlet louvers. The free area of the inlet louvers is 80 per cent of the inlet duct area. The actual friction-pressure losses in the transformation piece joining the inlet duct to the fan may be taken as 0.08 in. of water. The discharge duct is rectangular in cross section, 15 ft in length, has no elbows, and is the same size as the fan outlet. All duct work is of galvanized sheet steel. The static air pressure in the kiln is 0.15 in. of water. Saturated steam will be supplied to the heater at 227 F. (*a*) Select a non-ferrous blast heater for this installation. (*b*) Find the sizes of the ducts necessary. (*c*) Select a fan for the service to be rendered. (*d*) What horsepower motor is required if the drive between the motor and the fan shaft is 96 per cent efficient?

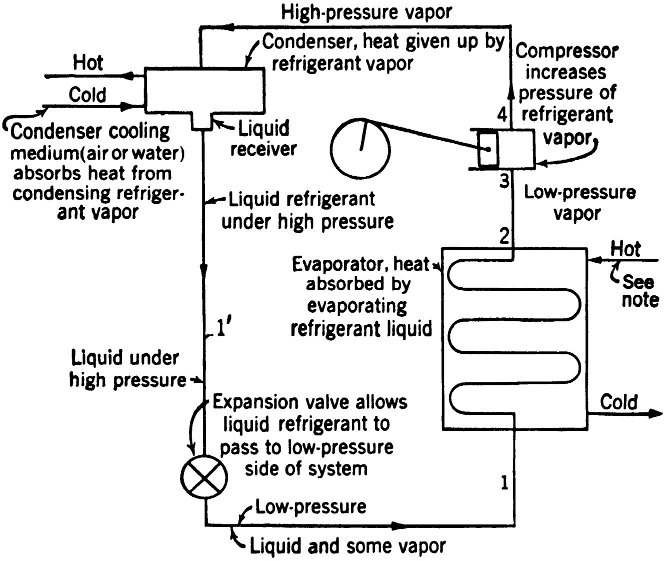
CHAPTER 16

MECHANICAL REFRIGERATION APPLICATIONS TO COOLING AND HEATING

358. Types of Mechanical Refrigeration Systems. Several different systems of mechanical refrigeration are used in present-day air-conditioning practice. The more commonly used types, which will be discussed in this chapter are: the compression system, the absorption system, and the steam-jet system. The first system mentioned is extensively used for summer cooling and is now being adapted to winter heating in an arrangement which is popularly called the "heat pump."

359. Fundamentals of a Compression Refrigeration System. All fluids behave in a manner similar to water in that they will evaporate if in the liquid state when heat is added or condense if in the vapor state when heat is taken away, at a temperature called the saturation temperature. The saturation temperature for any fluid depends on the pressure extent in the containing tube or vessel. Any fluid for which the relationship between saturation temperature and vapor pressure is suitable may serve as the working medium in a compression refrigeration system.

Figure 298 is a flow diagram for a compression refrigeration system. All the principal parts are labeled, and the path of the refrigerant is indicated. The refrigerant passes through a circuit which is divided into a high-pressure portion and a low-pressure portion. The pressure is maintained at different levels in the two parts of the system by the expansion valve or a capillary tube at one point and by the compressor at another. The function of the expansion valve is to allow the liquid refrigerant under high pressure to pass at a controlled rate into the low-pressure part of the system. Some of the liquid evaporates the instant that it passes the expansion valve, but the greater portion is vaporized in the evaporator at the low pressure which is maintained by the exhausting action of the compressor. In evaporating, each pound of liquid refrigerant absorbs its latent heat of vaporization, the greater part of which is conducted to it through the evaporator tubes from the air, water, or other material which is being cooled. The function of the compressor is to increase the pressure of the refrigerant vapor and discharge it into the condenser. Because of the high pressure maintained in the condenser the vapor condenses at a comparatively high temperature. In passing through the condenser the refrigerant gives up the heat which it absorbed in the evaporator plus the heat equivalent of the work done upon it by the compressor. This heat is transferred through tubes to the air or water which is used as the condenser cooling medium.


High-pressure vapor
Condenser, heat given up by refrigerant vapor
Hot
Cold
Condenser cooling medium(air or water) absorbs heat from condensing refrigerant vapor
Liquid receiver
Compressor increases pressure of refrigerant vapor
4
3
Low-pressure vapor
Liquid refrigerant under high pressure
2
Hot
See note
Evaporator, heat absorbed by evaporating refrigerant liquid
1'
Liquid under high pressure
Cold
Expansion valve allows liquid refrigerant to pass to low-pressure side of system
1
Low-pressure
Liquid and some vapor

Note: Medium to be cooled may be the air that is conditioned or it may be water or brine to be used in turn as a cooling agent

FIG. 298. Flow diagram for a compression refrigeration plant.

It may be said that this system serves as a heat pump since it is capable of absorbing heat at a low temperature level and discharging it at a high temperature level, just as a pump is capable of taking in a liquid at a low pressure level and discharging it at a high pressure level. The compression refrigeration system is analogous to an ordinary pump in that the amount of energy required to transfer a given amount of heat from the low temperature level in the evaporator to the high temperature level in the condenser is proportional to the magnitude of these two temperature levels on the absolute temperature scale.

360. Coefficient of Performance. Since the purpose of the system is to transfer heat from a low temperature level to a high temperature level through the expenditure of energy in the form of power, and, since the amount of heat transferred is usually greater than the heat equivalent of the power supplied, the performance of such a system cannot logically be expressed as an efficiency, so the term coefficient of performance has been substituted.

The coefficient of performance may be defined as the ratio of the heat absorbed by the refrigerant as it passes through the evaporator to the heat equivalent of the energy supplied to the compressor.

$$COP = \frac{\text{Heat absorbed in evaporator, Btu per hour}}{\text{Horsepower supplied to compressor} \times 2545}$$

In terms of the enthalpies of the circulating refrigerant at the critical points in the cycle (assuming no compressor-cylinder heat losses)

$$COP = \frac{h_2 - h_1}{h_4 - h_3} \tag{142}$$

where h_1 = enthalpy of the refrigerant entering the evaporator, Btu per lb.

h_2 = enthalpy of the refrigerant leaving the evaporator, Btu per lb.

h_3 = enthalpy of the refrigerant entering the compressor, Btu per lb.

h_4 = enthalpy of the refrigerant leaving the compressor, Btu per lb.

The enthalpy h_3 of the refrigerant as it enters the compressor will ordinarily be practically the same as the enthalpy h_2 as it leaves the evaporator, provided that the pipe line between these two points in the system is well insulated. Similarly the enthalpy h_1 will usually be nearly equal to the enthalpy of the liquid leaving the condenser and may be determined more conveniently from measurements taken before the expansion valve at 1′, Fig. 298, provided that the pipe line between the expansion valve and the evaporator is well insulated. The temperature of the refrigerant decreases abruptly from a temperature that is approximately that of the condenser to a temperature that is approximately that of the evaporator as it passes through the expansion valve. However, this phenomenon produces no change in enthalpy as the enthalpy of the liquid is decreased by exactly the amount that the enthalpy of vaporization is increased when a small portion of the liquid flashes into vapor.

361. Method of Rating Refrigeration Machines. The unit of capacity generally used in rating refrigeration machines is the ton.

A refrigerating machine is said to have developed a ton of refrigeration when its circulating medium has absorbed 288,000 Btu in the evaparator. This is the amount of heat that would have to be absorbed from one ton of water at 32 F in converting it to the same weight of ice. Refrigerating units are usually rated in terms of their capacity in tons per 24 hr; i.e., a machine having a rated capacity of one ton would be capable of absorbing 288,000 Btu in 24 hr. This rate of heat absorption is equivalent to 12,000 Btu per hr or 200 Btu per min.

The capacity which any compression-type refrigerating machine may develop is affected by the pressures which the compressor must maintain at its suction and discharge openings when the system is in operation. The standard rating of a refrigerating machine using a liquefiable vapor is the number of tons it delivers under standard conditions, which are defined as the respective pressures corresponding to saturation vapor temperatures of 5 F at the compressor intake and 86 F at the compressor discharge. It is further stipulated that the liquid entering the expansion valve shall be subcooled 9 deg F and that the vapor shall be superheated 9 deg F as it reaches the suction side of the compressor. The actual pressures which constitute standard conditions for any particular refrigerant may be obtained by referring to a suitable table giving the properties of that fluid. These pressures are given for several refrigerants in Table 110.

If actual operating conditions require a pressure at the compressor outlet that is higher than standard or a pressure at the compressor inlet that is lower than standard, the unit cannot be expected to develop its standard rating. The capacity of a compression refrigeration machine using an air-cooled condenser may be materially reduced during hot weather because the compressor is then called upon to maintain a pressure at its discharge which is considerably higher than standard. Conversely, a machine may develop a capacity exceeding its standard rating if the pressure differential required under operating conditions is less than that under standard conditions. A manufacturer of refrigerating equipment usually assigns several ratings to each unit in its line in addition to the standard rating. The applicable rating for any particular unit under consideration would then depend on the conditions under which it would operate in the proposed installation. Table 111 is an excerpt from a table showing several ratings and the corresponding required brake horsepower for a 4 by 4 in. ammonia compressor operating at 400 rpm.

362. Refrigerants for Air Cooling. Quite a number of media can be used as refrigerants, but those most practical for air conditioning are relatively few in number. Possible refrigerants are: water, H_2O;

TABLE 110

Properties of Refrigerants*

Properties	Water, H_2O	Carbon Dioxide, CO_2	Ammonia, NH_3	Sulfur Dioxide, SO_2	Freon F-12, CCl_2F_2	Carrene Number 2 or Freon F-11, CCl_3F	Methyl Chloride, Ch_3Cl
Latent heat of vaporization at 35 F, Btu per lb	1074.1	98.8	540.5	160.84	66.27	81.63	173.31
Specific volume of saturated vapor at 5 F, cu ft per lb	11.530	0.266	8.15	6.42	1.485	12.27	4.47
Specific volume of saturated vapor at 86 F, cu ft per lb	527.3	0.0479	1.77	1.185	0.389	2.24	1.08
Pressure of saturated vapor at 5 F, psia	0.024	331.95	34.27	11.80	26.51	2.93	21.15
Pressure of saturated vapor at 86 F, psia	0.615	1043.0	169.2	66.45	107.9	18.28	94.70
Difference between above pressures	0.591	711.05	134.93	54.65	81.39	15.35	73.55
Standard cycle theoretical horsepower per ton	1.15	1.843	0.991	0.995	0.997	0.935	0.963
Boiling temperature at standard atmospheric pressure, F	212	−109.3	−28	+13.8	−21.6	74.7	−10.6
Freezing temperature, F	32	−69.9	−107.86	−98.9	−247.0	−168	−144
Critical pressure, psia	3206	1069.9	1651	1141.5	582	535	969.2
Saturation temperature at the critical pressure, F	705.4	87.8	271.2	314.8	232.7	388.4	286.9
Average specific heat of the liquid in the temperature range 5 F to 86 F	1	0.77	1.12	0.34	0.23	0.21	0.38
Specific gravity of liquid	1	1.56	0.684	1.357	1.48	1.468	1.002
Soluble in oil	No	No	No	No	Yes	Yes	Yes
Leak tendency	Air in	Refrigerant out	Refrigerant out	Refrigerant out	Refrigerant out	Air in	Refrigerant out
Leak detector	Air in System	Orsat	Sulfur stick	Ammonium sulfite	Halide torch	Air in system	Soap bubbles
Toxic	No	No	Yes	Yes	Slightly	Slightly	Slightly
Explosive	No	No	Yes	No	No	No	Yes
Irritating	No	No	Yes	Yes	No	No	No
Corrosive	No	No	Yes to copper	Yes in presence of water	No	No	Slightly with aluminum

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* Data taken with permission from *Refrigeration Data Book*.

TABLE 111

REFRIGERATING CAPACITY DEVELOPED AND BRAKE HORSEPOWER REQUIRED BY A 4 BY 4 IN. TWIN-CYLINDER AMMONIA COMPRESSOR OPERATING AT 400 RPM*

Discharge Pressure Psi Gage and Corresponding Temperature, F	Capacity and Horsepower	Suction Pressure Psi Gage and Corresponding Temperature, F						
		5 − 17.2 F	10 − 8.4 F	15.7 0 F	20 + 5.5 F	22.5 + 11.3 F	25 + 11.3 F	30 + 16.6 F
95 psi	Tons	2.82	3.73	4.77	5.58	6.04	6.50	7.49
+ 61.1 F	Bhp	5.72	5.98	6.26	6.29	6.30	6.31	6.28
115 psi	Tons	2.65	3.50	4.49	5.27	5.72	6.17	7.12
+ 70.4 F	Bhp	6.04	6.41	6.75	6.88	6.95	7.00	7.03
135 psi	Tons	2.48	3.29	4.24	5.0	5.43	5.87	6 77
+ 78.7 F	Bhp	6.30	6.76	7.21	7.39	7.48	7.58	7.68
155 psi	Tons	2.33	3.12	4.03	4.75†	5.17	5.59	6.46
86.15 F	Bhp	6.48	7.07	7.59	7.87	8.03	8.10	8.28
165 psi	Tons	2.25	3.04	3.93	4 64	5.05	5.46	6.31
89.65 F	Bhp	6.55	7.17	7.77	8.10	8.21	8.37	8.52
175 psi	Tons	2.19	2.95	3.83	4.52	4 93	5.34	6.17
93.05 F	Bhp	6.71	7.30	7.93	8.28	8 43	8.55	8.75
185 psi	Tons	2.01	2.88	3.74	4.42	4.82	5.23	6.04
96.25 F	Bhp	6.81	7.40	8.07	8.46	8.61	8.78	9 01

* Vilter Manufacturing Company.
† Standard rating.

carbon dioxide, CO_2; ammonia, NH_3; sulfur dioxide, SO_2; dichlorodifluoromethane, CCl_2F_2, commonly known as Freon, F-12; monofluorotrichloromethane, CCl_3F, commonly called F-11 and Carrene number 2; and methyl chloride, CH_3Cl.

Several significant physical and thermal properties of the above-mentioned refrigerants are given in Table 110.

It may be observed from the data of Table 110 that the pressures required for water vapor in the standard temperature range are very low and that the pressures for carbon dioxide in the same range are very high. All the other refrigerants listed require pressures which are more easily obtained.

The weight of refrigerant which must be circulated per ton of refrigeration produced depends principally upon the latent heat of vaporization of the refrigerant used. The volume of gas which must be compressed per pound of refrigerant circulated depends upon its specific volume as it enters the compressor cylinder. The work required in the compressor cylinder per cubic foot of vapor entering is proportional to the difference between the pressure of saturated vapor at 86 F and the same property at 5 F when the system is operated under standard conditions. The standard-cycle theoretical horsepower required per ton of refrigeration produced is the best measure of the efficiency of the refrigerant and is affected by all three of the properties which have just been mentioned. It may be noted that when carbon dioxide and water are eliminated from consideration, the power requirements are approximately the same for all the other refrigerants included in the preceding list. The table also includes several additional information items in regard to each of the refrigerants.

Water is inexpensive, non-toxic to the human body, and is easily controlled. However, large volumes of the vapor must be handled at extremely low pressures in certain portions of the system. Special apparatus for the employment of water as a working medium to produce a refrigerating effect in air-conditioning systems is described in Art. 365.

Because of the extremely high pressures required and the expensive compressor and piping necessary to handle them, carbon dioxide is being used less and less as a refrigerating medium.

Ammonia is a common refrigerant in ice-making and cold-storage plants. It has a higher latent heat of vaporization than any other refrigerant except water and has several other desirable properties, but, because of its bad effects on the human body if it escapes from the system, it is seldom used in air-conditioning work.

Sulfur dioxide is still used in many household refrigerators, but, principally because it is an irritant, most manufacturers of such equipment now prefer to use one of the newer refrigerants. It is non-explosive and has some lubricating properties when in the liquid phase.

Freon-12 is a refrigerant which can be handled in a satisfactory manner by any type or size of reciprocating compressor. It is considered as a safe refrigerant although some of its decomposition products when it is subjected to high temperature are poisonous. Its principal disadvantages are its low latent heat of vaporization and the high cost of producing it.

Carrene number 2, sometimes called F-11, has only slightly toxic effects on the human body, and its vapor can be readily handled by centrifugal compressors because of the low pressure required in the condenser.

Methyl chloride though classed as an explosive is not readily ignited. It produces some toxic effects but necessitates only a moderately high pressure in the condenser and is well adapted to use in systems employing small reciprocating compressors. It possesses good pressure, volume, and thermodynamic characteristics, but it is seldom used in large-capacity units because of safety considerations.

Tables giving data useful in the design or operation of refrigerating systems for all the refrigerants which have been discussed plus certain others may be found in the *Refrigerating Data Book.*[1]

363. The Absorption Refrigeration System. Figure 299 shows a simplified flow diagram for an absorption refrigeration system. The principle of this system is identical with that of the compression system except in the manner in which the pressure of the refrigerant vapor is increased prior to its condensation in the condenser. The compressor of the compression system is replaced by the absorber, the pump, and the generator in the absorption system. Instead of compressing the low-pressure refrigerant vapor from the evaporator pressure to the necessary condenser pressure, it is first absorbed by a weak solution of the refrigerant in water. The strong solution thus formed is then pumped into the generator where it is heated. In the heating process the refrigerant vapor is driven out of the solution and conducted to the condenser under the necessary pressure.

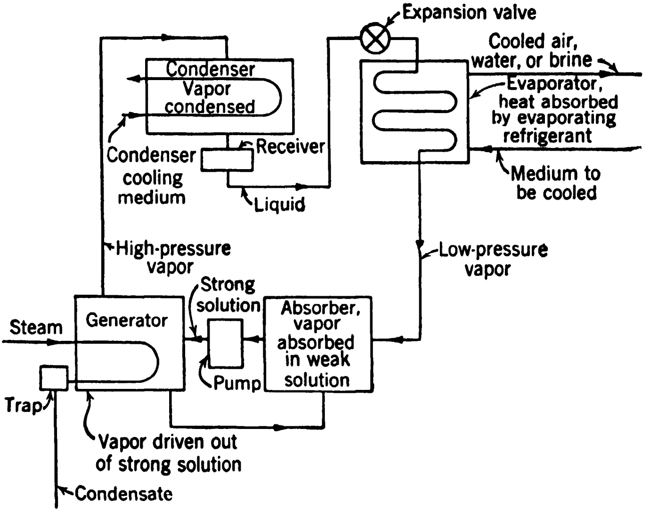

FIG. 299. Flow diagram for an absorption refrigeration plant.

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[1] *Refrigerating Data Book,* American Society of Refrigerating Engineers, 50 West 40th Street, New York, New York.

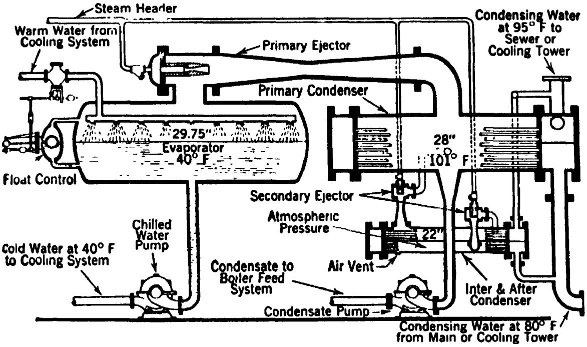
Considerably more apparatus is required for the operation of the absorption system, but the cost of operation may be lower than that of a comparable compression system where exhaust steam or some other low-cost heating medium is available for use in the generator. Inasmuch as a given volume of the solution will absorb many times that volume of vapor, the volume handled by the pump of the absorption system is small compared to the volume handled by the compressor of the compression system, and the power costs are proportionately lower. Ammonia is the refrigerant best suited to this system.

364. Water Vapor Refrigeration. Thus far the refrigeration cycles considered have involved putting the gaseous refrigerant under increased pressure, condensing it, and then dropping its pressure so that vaporization of the refrigerant will occur with the consequent absorption of heat from the fluid (either water, brine, or air) cooled. Water-vapor refrigeration is also a compression cycle in which a portion of the water handled is made to vaporize at extremely low absolute pressures. In such operation the unvaporized portion of the water loses sensible heat, and the vaporized portion of the water gains latent heat in the chamber where evaporation takes place. As a result of this heat transfer the water not vaporized has its temperature decreased.

Steam-jet and centrifugal compressors are devices used in the production of the low-pressure region in which flashing of water into vapor occurs. These appliances also remove from the evaporating chamber the large volumes of water vapor formed together with its entrained air and other gases and put the vapor-gas mixture under sufficient pressure so that it will flow to the condenser. The vacuum required for the vaporization of the water ranges from 29.5 to 29.75 in. of mercury referred to a 30-in. barometer and is dependent upon the final water temperature which is to be secured.

365. Steam-Jet Refrigeration Apparatus. Commercial equipment for the production of refrigeration by means of steam jets, using water as the refrigerant, is shown in Fig. 300. Steam passing through the nozzles of the primary ejector acquires a high velocity and aspirates the vapor and gases from the evaporator chamber to produce a low pressure within it. The warmed water, returning from air washer or cooling unit, is broken into a spray in the evaporator chamber and a small part of it, usually less than one per cent, flashes into vapor. The remainder of the water is cooled, losing an amount of sensible heat equal to the latent heat taken up by the vapor. As a result it is possible to cool water to 40 F if a vacuum of 29.75 in. of mercury, referred to a 30-in. barometer, can be maintained in the evaporating chamber.

The steam passing through the primary ejector and the vapor taken from the flash chamber or evaporator are condensed in the primary condenser, in which a vacuum is also maintained. In order to maintain a sufficiently low pressure in the primary condenser a secondary steam ejector is used to remove the air and non-condensable gases from it. The secondary ejector is served by the water-cooled intercondenser which reduces to water most of the steam and vapor handled by the first secondary ejector. Any uncondensed vapor and the non-condensable gases removed from the evaporator chamber escape to the atmosphere.


Steam Header
Warm Water from Cooling System
Primary Ejector
Condensing Water at 95° F to Sewer or Cooling Tower
Primary Condenser
29.75"
Evaporator 40° F
28"
101° F
Float Control
Secondary Ejector
Atmospheric Pressure
22"
Chilled Water Pump
Cold Water at 40° F to Cooling System
Condensate to Boiler Feed System
Air Vent
Inter & After Condenser
Condensate Pump
Condensing Water at 80° F from Main or Cooling Tower

(*American Blower Corp.*)

FIG. 300. Ross decalorator.

Steam at any pressure above 12 to 15 psig may be used in the ejectors. The higher the initial steam pressure the less the steam requirements per unit of refrigeration. Figure 301 gives data[2] relative to the steam consumption of steam-jet refrigeration units. An initial steam pressure of 125 psig gives about the best operating results with steam-jet refrigeration as far as the amount of steam required per unit of refrigeration is concerned. Pressures above 125 lb produce little reduction in the steam consumption per ton of refrigeration. Reduction of the initial steam pressure below 125 psig requires a greater amount of steam to produce a given refrigerating effect. Thus with an initial steam pressure of 50 psig the amount necessary will be approximately 1.14 times that required with 125 psig and that with 10 psig 1.66 times as much. The lowest practical steam pressure is in the neighborhood of 2 psig.

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[2] "Refrigeration by Steam Jet," by D. W. R. Morgan, *Power*, p. 506, September 1934.

Steam-jet refrigeration is not well adapted to units of less than 6 tons capacity. Considerable cooling water is required for condensing the steam used, the amount per ton of refrigeration being dependent upon the size of the machine and the initial temperature of the water as it enters the condensers.

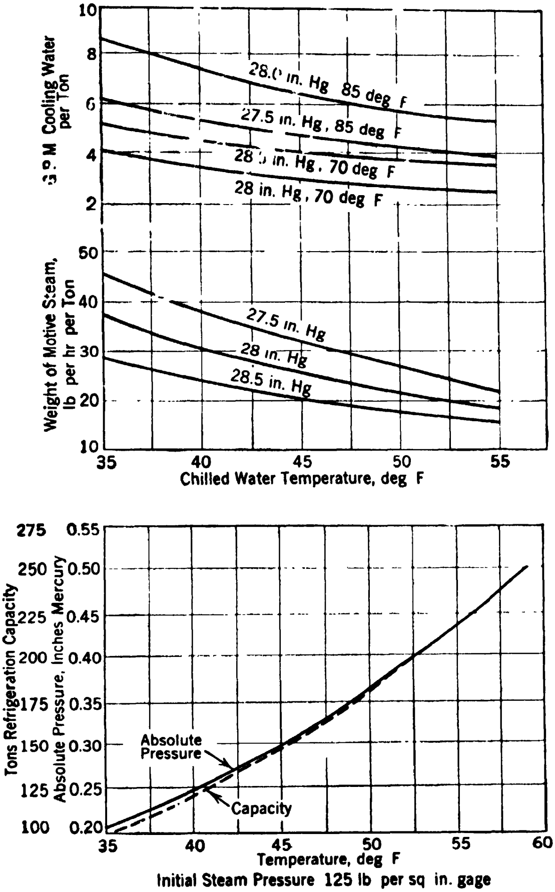
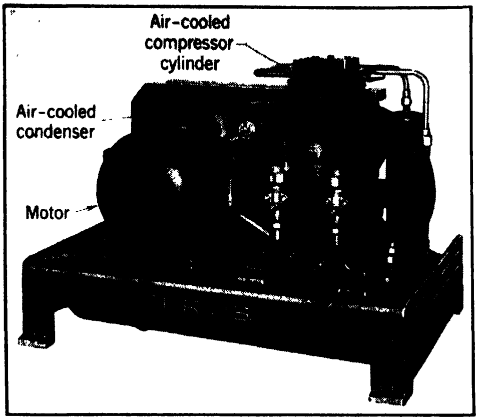


FIG. 301. Steam and cooling water consumption and capacity of a steam-jet refrigeration unit.

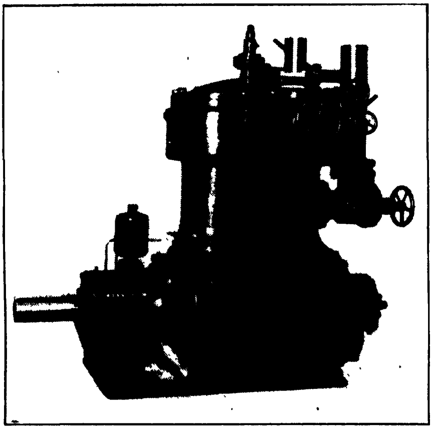
366. Reciprocating Compressors. Reciprocating compressors are made in many different sizes to serve compression refrigeration systems with capacities varying from $\frac{1}{2}$ ton to 100 tons. This type of compressor is invariably used with refrigerants such as ammonia and Freon-12 with which a high pressure is required in the condenser. Single-cylinder and multi-cylinder units are available, and the majority of present-day designs are arranged to be driven by electric motors through multiple *V* belts. Small units of 5-ton capacity or less may be arranged integrally with an air-cooled condenser, fan blades on the compressor flywheel providing the required air circulation over the condenser tubes. Figure 302 shows a 3-hp unit with integral air-cooled condenser, and Fig. 303 shows a two-cylinder water-cooled ammonia compressor rated at 100 tons under standard conditions. As a general rule approximately one horsepower of motor capacity for driving the compressor is required for each ton of refrigerating capacity of the system.



(*Curtis Manufacturing Co.*)

Fig. 302. Self-contained motor-driven refrigeration unit with air-cooled compressor and condenser.

367. Centrifugal Compressors. This type of compressor is well adapted to the handling of refrigerants such as Carrene number 2, where the volume passing through the compressor is comparatively large and the required increase in pressure is comparatively small. One advantage of the centrifugal compressor is that its high rotative speed makes it suitable for direct connection to electric motors or steam turbines, thereby simplifying the drive and effecting a saving in floor space. Another advantage is that there are no piston rings or valves to wear out so that maintenance of this type of compressor would, in general, be less costly.

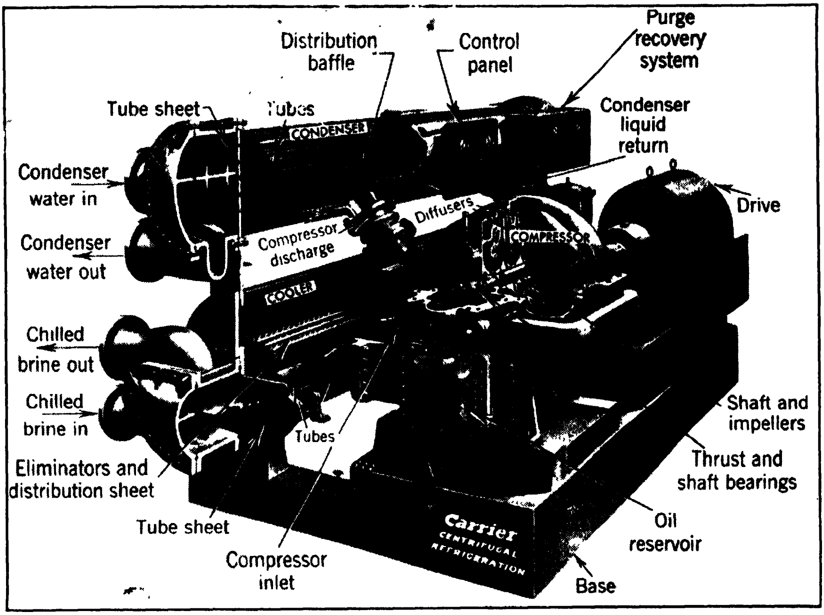


(*Vilter Manufacturing Co.*)

Fig. 303. Two-cylinder, water-cooled, 100-ton capacity ammonia compressor.

The efficiency of compression is usually lower with the centrifugal compressor than with the reciprocating type, but this disadvantage may be more than offset by the advantages which have been previously discussed.

Figure 304 shows the arrangement of a centrifugal compressor for use with Carrene number 2. The operation of the compressor under standard conditions produces a vacuum of 24 in. of mercury in the cooler where liquid Carrene is allowed to fall from the distributor plate over tubes. A centrifugal pump lifts the liquid refrigerant from the base of the cooler to the space above the distributor plate. At the low pressure within the cooler chamber the refrigerant boils and abstracts its latent heat from either water or brine flowing through the cooler tubes. The refrigerant vapor formed passes between eliminator plates to remove slugs of liquid. The gas then passes through the compressor where its pressure is increased to 3.58 psig. At this pressure the gaseous refrigerant enters the condenser where it is reduced to a liquid by water passing through the condenser tubes. The heat taken from the Carrene in the condenser is carried away by the condenser cooling water. The liquid Carrene falls to the space at the bottom of the condenser, and from there it passes through a float trap into the cooler where the cycle begins again.

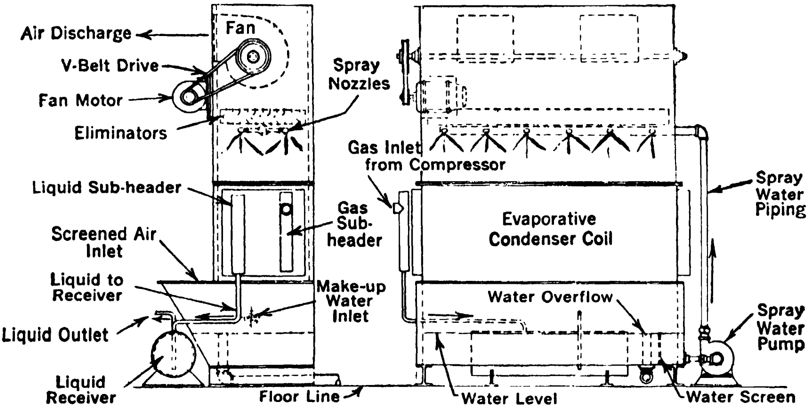


(Carrier Corp.)

FIG. 304. Centrifugal-compressor refrigeration unit.

368. Refrigerating Machine Condensers. These parts of a refrigerating system are of great importance as it is within them that the heat acquired by the refrigerant in the system is disposed of at a comparatively high temperature level.

In general, condensers may be divided into two groups, those cooled by air and those cooled by the action of water. Air-cooled condensers are employed only with small units, outside air being forced over the cooling tubes by the action of a fan or blower. Air-cooled condensers usually cause the refrigerant to discharge its heat at a higher temperature level than water-cooled condensers. Consequently the power requirements per ton of refrigeration are greater with air-cooled units than with water-cooled units.

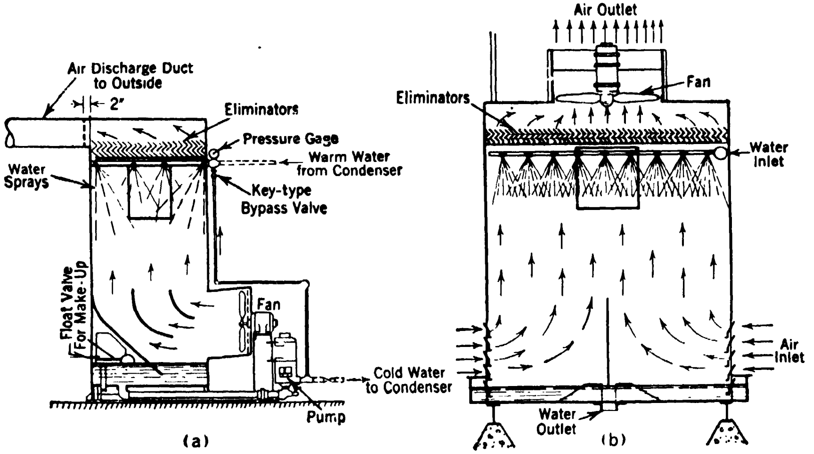


(*The Trane Co.*)

FIG. 305. Evaporative condenser.

Water-cooled condensers are built as shell and tube units with one or more water passes, double-tube condensers, and tubular units upon which the cooling water is sprayed. The condenser of Fig. 304 is a shell and tube type in which the refrigerant to be cooled and liquefied is confined within the shell surrounding the tubes through which water is circulated. Double-tube condensers have inner tubes through which water is circulated and outer tubes surrounding the inner tubes, the refrigerant being confined in the spaces between them.

Evaporative condensers, Fig. 305, have considerable use. They may be installed either indoors or outdoors. The action of an evaporative condenser is secured by spraying water upon the tubes of a refrigerant cooling coil located within the unit housing. The water trickles over the tubes in a direction opposite to the flow of air produced by a fan. Cooling of the water occurs as some of it vaporizes in the air stream. The water falls to a bottom reservoir from which it is taken by a pump and discharged through the sprays. The condensed refrigerant collects in a receiver at the bottom of the apparatus. The cooling water is recirculated, and about 10 per cent of the water handled is evaporated to cool the water and thereby condense the refrigerant. This results in a saving of about 90 per cent of the water that otherwise would be required for condensing purposes. When an interior installation is made a duct is necessary to convey cooling air from outdoors to the unit, and an exhaust duct is required to carry away the heated and humidified air.



(a) Indoor

(b) Outdoor

(*Binks Manufacturing Co.*)

FIG. 306. Cooling towers.

369. Methods of Reducing the Temperature of Recirculated Condenser Cooling Water. Cooling by water in condensers is a very effective method of disposing of heat put into the refrigerant during the cycle. In large plants a considerable quantity of condensing water may be required. The cost of this water may be a rather large item in the expenses of operation if it is discharged to the sewer. Lack of an adequate water supply and prohibition of drawing large quantities of it from city mains and also of discharging it to sewers have led to the introduction of methods of cooling and reusing condensing water. The principles of the methods of reducing the temperature of condensing water before using it again are essentially the same as those of power plants where the steam condenser water is used again and again. One method is spraying the water into the air where it loses some heat by contact with the air but more largely by the evaporation of some of the water sprayed. The water may fall from the sprays into a pond from which it is withdrawn as needed. The spray pond is not practical with most buildings in cities because of the lack of space. The commonest alternative is some form of a cooling tower located upon the building roof. The water is sprayed into a tower filled with latticework or other filling over which the water trickles. Air either by gravity flow or by the action of a fan is forced upward through the filling, and evaporative cooling takes place as some of the water handled is evaporated. It is, of course, necessary to supply the small amount of make-up water from city mains to replace that evaporated. Small units embodying many of the features of an air washer may be installed either indoors or outdoors with outside air circulated by means of a fan through the spray chamber. Such units are shown by Fig. 306.

The principles of adiabatic saturation apply to water cooled by means of sprays; theoretically the lowest temperature to which the water may thus be reduced is the wet-bulb temperature of the air coming in contact with it. As in adiabatic saturation the efficiency E obtained is

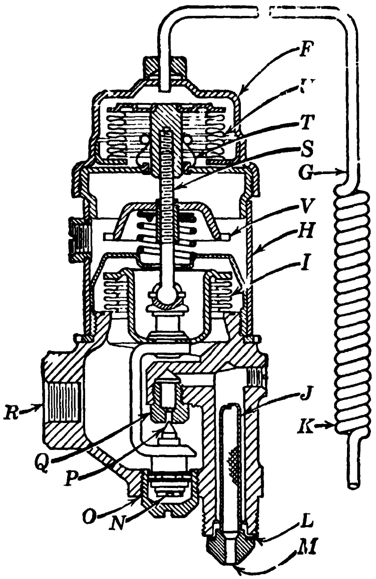
$$E = \frac{t_1 - t_2}{t_1 - t'} \times 100 \qquad (143)$$

where t_1 = temperature of the water entering the cooling tower, deg F
t_2 = temperature of the water leaving the tower, deg F
t' = wet-bulb temperature of the air, deg F

The cooling efficiencies may range from 50 to 75 per cent.

370. Expansion Valves. The expansion valve is a small but very important part of either a compression or an absorption system. Its purpose is to allow the liquid refrigerant to pass into the evaporator at the proper rate. Full utilization of the heat-transfer surface requires the presence of liquid refrigerant in all parts of all of the tubes. However, it is important that no liquid be carried out of the evaporator with the vapor that is returned to the compressor; it is therefore the usual practice to maintain a few degrees of superheat in the refrigerant at this point in its circuit.

Figure 307 is a sectional view of a thermostatic expansion valve which is designed to automatically regulate the flow of liquid refrigerant into the evaporator so that the vapor leaving is always superheated approximately 10 F. The liquid refrigerant enters at M and passes through the strainer J to the orifice Q. The needle valve P is attached to a yoke member which is rigidly attached to the bellows *I* in the valve body. When the pressure in the body, which is the same as the pressure in the suction line to the compressor, increases, it actuates this bellows and pulls the valve against its seat. A decrease in this pressure tends to open the valve. The power element *F* contains a bellows *U* which is attached by means of the capillary tube *G* to the feeler bulb *K* which may be attached to the compressor suction pipe at the evaporator outlet. The feeler bulb is charged with refrigerant so that the pressure exerted on the bellows *U* is determined by its temperature.



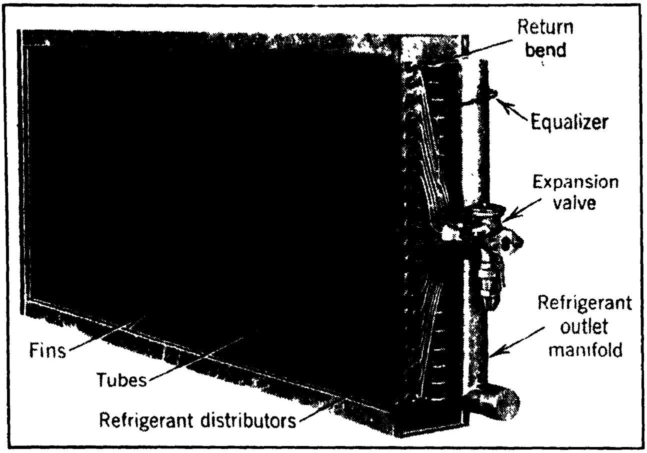
F. Power element
G. Capillary tube
H. Stainless-steel extension
I. Body bellows
J. Inlet strainer
K. Feeler bulb
L. Strainer gasket
M. Inlet connection
N. Needle swivel
O. Plug, hermetically sealed
P. Needle
Q. Seat
R. Outlet connection
S. Push rod
T. Anti-chatter device
U. Power element bellows
V. Factory adjustment

(Detroit Lubricator Co.)

FIG. 307. Thermostatic expansion valve.

The bellows *U* works against the bellows *I* through the push rod *S* so that an increase in the temperature of the feeler bulb tends to cause the needle valve to open. Since the pressure in the valve body and in the bellows *I* is determined by the evaporating temperature and the pressure in the bellows *U* by the temperature of the vapor leaving the evaporator, the operation of the expansion valve is controlled by the difference between the two or the number of degrees of superheat. Tests have indicated that a thermostatic expansion valve of this type will maintain practically a constant number of degrees of superheat in the vapor leaving the evaporator over a wide range of operating pressures.

371. Direct Expansion Coils. Figure 308 shows a type of evaporator known as a direct expansion coil in which the refrigerant is used to cool the air directly. Use of a coil of this type results in a less complex air-cooling and dehumidifying system than is the case when the air is cooled with chilled water which in turn must be passed through another heat exchanger which transfers heat from it to the evaporating refrigerant. The coil shown in Fig. 308 is the preferred construction in which each section is separately fed with liquid refrigerant from a manifold by means of a small tube. The liquid refrigerant is fed into the tubes through distributors at a controlled rate by the expansion valve.



(*Kennard Corp.*)

FIG. 308. Direct-expansion coil.

372. Heat Exchangers for Preparing Chilled Water. Although the system is necessarily more complicated when chilled water is used as the air-cooling medium instead of the refrigerant, it is frequently advantageous to use it in large buildings where it is necessary to provide several separate air-conditioning units. It is often possible to provide a compressor, a condenser, and a combination evaporator and water cooler, each with sufficient capacity to handle the entire cooling load of the building (see Fig. 304), The chilled water furnished by such a unit is then distributed through insulated pipes to the several cooling units located in different parts of the building. The employment of this overall scheme results in the centralization of all the equipment which handles the refrigerant, thereby reducing the initial investment, saving floor space, reducing maintenance charges, and reducing to the absolute minimum the number of joints in the refrigerant circuit and thus the chances for loss of costly refrigerant. When a toxic refrigerant such as ammonia is used, safety regulations do not permit its evaporation in a direct-expansion coil.

373. Reversed Refrigeration Used for Heating. Article 359 in brief indicates that a refrigeration system is in effect a heat pump which can absorb heat at a low temperature level and discharge it at a higher temperature level. Systems are used extensively for pumping heat from refrigerated materials or spaces and discharging it to any available cooling medium such as water or air. They can also be used for pumping heat from any available heat source, such as the outdoor air, the ground, or the water in a well, and discharging it to air or water at a comparatively high temperature which in turn may be used for heating buildings during cold weather or heating water for service requirements.

Because the heat transferred from the heat source to the heating medium may be a much greater amount than the heat equivalent of the power required by the compressor, the cost of operating such a system for the purpose of heating may not exceed the cost of operating systems in which heat is derived from the burning of fuels. In localities where electric power is obtainable at less than one cent per kilowatthour the cost of heating by means of reversed mechanical refrigeration may be less than the cost of operating any type of conventional heating system.

At the present time the initial cost of a compression refrigeration system arranged for heating is considerably greater than the cost of any conventional heating system of comparable capacity. However, a refrigeration system properly installed may be used for summer cooling as well as for winter heating, and the required outlay for such a unit may be less than the combined cost of a conventional heating plant and a conventional summer-cooling system.

374. Possible Apparatus Arrangements. Figure 309 is a diagram of the simplest type of reversible compression refrigeration system in which the source of heat for winter heating is the outdoor air. The damper positions shown are for winter heating. Fan 1 draws outside air over the evaporator where it gives up heat which is transferred through the metal tubes to the evaporating refrigerant liquid. This air is then discharged back to the outside at a lower temperature than that at which it entered the system. The low-temperature refrigerant vapor passes to the compressor where its pressure is raised to the necessary level such that it can be condensed in the condenser at a temperature that is considerably higher than that of the recirculated air. The dampers direct this air which is handled by fan 2 over the condenser tubes where it absorbs heat from the condensing refrigerant. The heat which the refrigerant gives up to the air as it passes through the condenser is equal to that which it absorbed in the evaporator plus the energy that was added due to the work done upon it by the compressor. The heated air from the condenser is delivered by fan 2 to the conditioned space and the liquid refrigerant passes through the expansion valve on its way to the evaporator and the completion of its circuit.

By turning all of the dampers 90 deg, which may be accomplished automatically by suitable thermostatic controls, the system is put in a position to cool the air delivered to the conditioned space instead of heating it. The operation of the refrigeration system is exactly the same when the dampers are positioned to provide summer cooling as when the apparatus is used for winter heating. When all the dampers indicated in Fig. 309 are turned 90 deg from the positions shown, the outside air passes over the condenser tubes where it absorbs heat from the condensing refrigerant, and the recirculated air from the conditioned space is cooled by the evaporating refrigerant liquid as it passes through the evaporator. With the dampers arranged for summer cooling, fan number 1 returns the heated air to the outside, and fan 2 delivers the cooled air to the conditioned space.

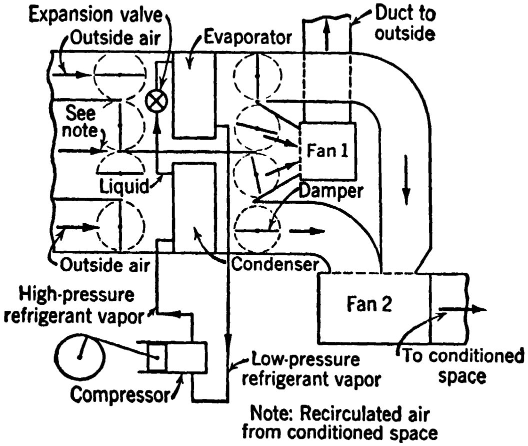
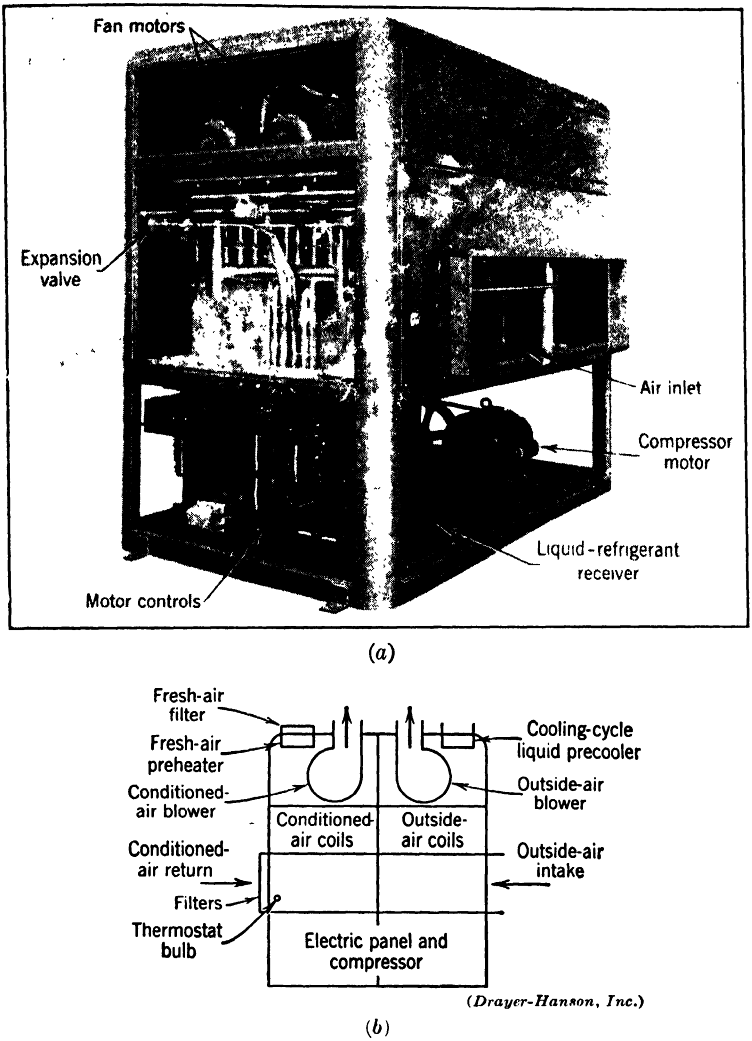


FIG. 309. Dual-purpose heat pump adjusted for winter heating.

Figure 310 illustrates a commercially available "packaged heat pump" designed to abstract heat from a stream of outdoor air when heating is required and to dissipate heat to the same medium when there is need for cooling and dehumidification. The unit shown in



(*Drayer-Hanson, Inc.*)

FIG. 310. Packaged heat pump operating on air to air basis. (*a*) Pump unit with enclosing panels removed. (*b*) Schematic elevation.

Fig. 310 is provided with two fans and contains a switching valve in the high-pressure vapor line and other necessary valves and piping details to permit interchanging the functions of the coil in the outdoor air and the coil in the conditioned air stream. The coil in the outdoor-air stream serves as the evaporator when heating is required and as the condenser when cooling is needed. In a similar manner the coil in the air stream from and to the conditioned space serves as the condenser during cold weather and as the evaporator during hot weather. This scheme eliminates the necessity of dampers and alternate air passages and facilitates the arrangement of all the apparatus in a housing of reasonable proportions.

The apparatus arrangements shown in Figs. 309 and 310 may constitute satisfactory heat pumps for use in a climate where the outdoor temperature never falls below 20 F. However, in localities where lower temperatures may prevail during the coldest weather, it is necessary to obtain heat from another source if the heating load is to be met without the installation of excessively expensive equipment.

Water may be used in place of outdoor air as a source of heat for winter heating and may also serve as a satisfactory cooling medium for the condenser during hot weather. The water may be taken from city mains, or from a well, and discharged into a sewer or drain, or it may be circulated through the casing of a deep well. Water may also be circulated through pipes buried in the ground. In any case a heat pump which uses water as a source of heat is necessarily more complex than one in which heat is obtained from, or discharged to, a stream of outdoor air. Figure 311 shows a possible layout of a system in which water circulated through a coil buried in the ground serves as the source of heat or the sink for waste heat, as the occasion demands. With this arrangement no change is required in the air-handling system when switching from winter heating to summer cooling, or vice versa. The dampers shown are for varying the proportion of fresh air, and

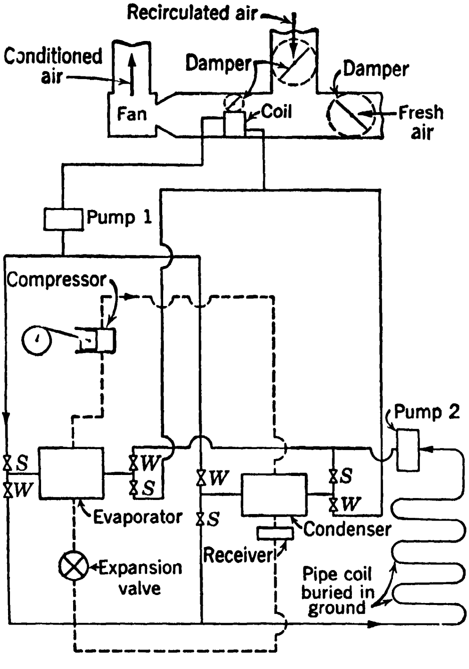


FIG. 311. Heat pump with ground coils through which water is circulated as an intermediate heat-transfer medium.

their settings are independent of the type of processing to be done. A coil by-pass is provided so that part of the air stream may be cooled sufficiently to produce dehumidification without delivering air to the conditioned space that is too cool for comfort. Likewise, the path of the refrigerant is not affected by the type of service performed by the system. Four valves labeled *S* and four others labeled *W* are provided for properly directing the flow of water to accomplish the desired purpose. The valves which are labeled *S* are to be open during summer weather and closed during the winter time. Similarly the valves labeled *W* are to be open in the winter and closed during the summer. During summer operation pump 2 circulates water from the buried coil through the condenser tubes and back to the coil. The water absorbs heat from the condensing refrigerant in passing through the condenser and loses it to the soil in passing through the ground coil. Pump 1 provides circulation in another closed circuit consisting of the evaporator and the coil in the air stream. In the overall processes heat is removed from the air that is being cooled by the water handled by pump 1, which transfers it to evaporating refrigerant in the evaporator. The refrigerant vapor then transfers the heat to the condenser where it is picked up by the water in the other water circuit which in turn transfers it to the ground through the buried coil. During winter operation pump 2 circulates water from the buried coil through the evaporator where it gives up heat to the evaporating refrigerant. The water circulated by pump 1 passes through the condenser where it absorbs heat from the refrigerant and transfers it to the air as it later passes through the coil which then serves as an air heater.

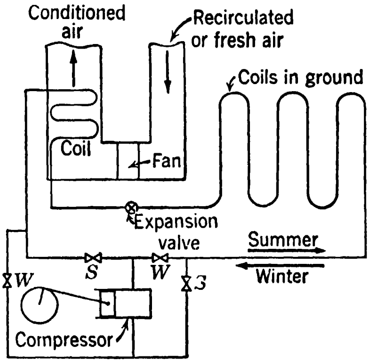


FIG. 312. Heat pump with a direct-expansion coil buried in the earth.

Another scheme for using the ground as the source of heat is illustrated in Fig. 312. When this arrangement is used for summer cooling, the coil, over which the air to be conditioned flows, is the evaporator in the refrigerant circuit. The compressor raises the pressure of the refrigerant vapor and forces it to flow through the pipe coil buried in the ground where condensation occurs. The heat given up in the process is absorbed by the surrounding soil. During this type of operation the two valves *S* are open and the two marked *W* are closed.

When heating is required by the space served, the position of each of the 4 valves is reversed causing the high-pressure refrigerant vapor to flow from the compressor through the coil in the air stream, which then serves as the condenser. The heat given up by the condensing refrigerant raises the temperature of the air which is being conditioned. The liquid refrigerant from the condensing coil in the air stream then passes through the expansion valve and evaporates in the buried coil, absorbing heat from the soil during the process. Heat is thereby transferred from the soil to the air which is being processed through the medium of the circulating refrigerant.

This system is considerably simpler than the one shown in Fig. 311 in which water is used as an intermediate heat-carrying medium. However, the system is subject to certain operating hazards not present in the other arrangement. For example, oil from the compressor may be trapped in the buried coil, or a leak developing in the external piping may result in the loss of a large part of the refrigerant before it is detected.

375. Comparison of Heat Sources. Outdoor air as a source of heat for winter heating permits the development of packaged units such as the one shown in Fig. 310. However, it does not appear to be economically feasible to rely upon this convenient heat source for "heat-pump" installations which are to serve buildings in locations where the winter weather may be severely cold.

The ground is the ultimate source (or sink) for all heat-pump installations which do not operate on an "air-to-air" basis. If water from a well or a city water system is used to absorb heat from the condenser or to supply heat to the evaporator, the initial temperature of the water will approximate the ground temperature. There is no fundamental difference in the possibilities of the various schemes in which the ground is used as the heat reservoir to give up heat when needed and to store heat when the air to be processed must be cooled. The scheme to be chosen for a particular installation will depend upon a number of circumstances, such as the policy of the local water company in regard to furnishing water for heat pumps, cost of city water if available for this use, condition of the local underground water reservoir, local cost of deep-well drilling, available adjacent areas for buried coils, and type of soil. Buried coils seem to be the most practicable method of making the ground the heat reservoir for a heat-pump installation in the majority of instances, but more information is needed about heat transfer from the ground to the circulating medium, and vice versa, covering a wide range of soil types. It should be pointed out that pipes buried in a shallow trench do not serve as effectively as the use of water from a well or the circulation of water through a double pipe immersed in a well because the temperature of the earth near its surface increases in summer and decreases in winter, though to a considerably lesser degree than is the case with the outdoor air. Figure 313 shows the variation in the soil temperature at different depths plotted from data[3] taken by Houghten and others during the year 1941.

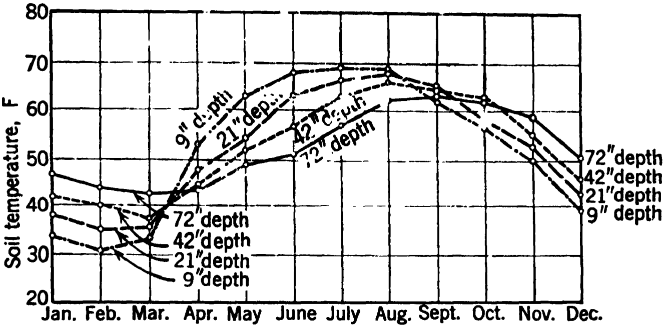


FIG. 313. Seasonal temperature variations at different depths in well-drained red clay soil at Pittsburgh, Pennsylvania, 1941.

376. Design of Heat-Pump Installations. If a packaged air-to-air unit is to be used the air-conditioning engineer should proceed as follows:

1. Secure weather data for the locality where the installation is to be made and decide on the summer and winter outdoor temperatures to be used for design purposes.
2. Estimate the heat gain for summer design conditions taking into consideration every possible interior source of heat (see Chap. 17).
3. Estimate the heat loss for winter design conditions (see Chap. 4).
4. Select a packaged unit guaranteed to carry both the summer cooling load and the winter heating load under the respective design conditions.
5. Design a system of ducts and registers to properly distribute the conditioned air and properly collect the air to be recirculated. The air volume to be handled and the allowed resistance of the system will be specified by the manufacturer of the packaged unit.

Packaged units using water from a private well or from a city water main may be practically as simple as units operating on the air-to-air basis if local conditions permit discharging the water to a drain. A packaged unit of this type would have a higher coefficient of performance than one using air as the source or sink. Use of packaged heat pumps of this type appears to be a practicable possibility in cities using surface water from large rivers or lakes. Though the temperature of the water in a river or a lake fluctuates considerably with changing seasons, the temperature of the water arriving at any building served by the system is rather stable throughout the year because of the modulating effect of the earth surrounding the underground distributing pipes.

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[3] "Heat Loss through Basement Walls and Floors," by F. C. Houghten, S. J. Taimuty, Carl Gutberlet, and C. J. Brown, *ASHVE Trans.*, Vol. 48, 1942.

If, because of climatic conditions, it is impractical to use air as the heat reservoir and if water in the quantities that would be needed may not be discharged to a drain, then it becomes necessary to provide either a deep well or a buried coil through which some suitable heat-carrying medium may be circulated to and from the heat-pump apparatus. It has previously been mentioned that insufficient data are available at the time of this writing, but some experience has been gained which may be used as a guide until more complete information has been obtained.

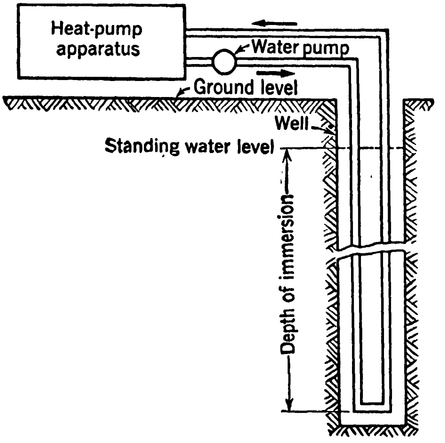

FIG. 314. Schematic diagram of a deep well as a source or sink for a heat pump.

Experience with a double pipe inserted in a well having a fluid depth of 200 ft indicates that such an arrangement can be depended upon to supply at least 28,000,000 Btu during a heating season. It appears likely that the capacity of the arrangement shown in Fig. 314 for supplying heat to a heat pump or for receiving heat from a heat pump would be directly proportional to the depth of immersion of the double pipe in the water standing in the well casing.

When buried pipes carrying a suitable medium are used to extract heat from the ground or to dissipate heat to it, the number of square feet of pipe surface required will depend upon the temperature of the ground, the temperature of the circulating medium, the overall coefficient of heat transfer from the circulating medium to the ground and the conductivity of the soil surrounding the pipes. The very meager data on the thermal conductivities of soils indicate that they vary from 1.0 Btu per hr per sq ft per deg F per in. of thickness for a dry, light soil to 16.0 for a wet subsoil. In general, the conductivity of dry soils increases with the density, and the conductivity of all soils increases with the moisture content up to the point of saturation. The density of the soil appears to have little effect on its thermal conductivity when it is saturated with water. Pipes should be placed at a depth below the ground surface which will insure reasonable freedom from seasonal temperature variation. The curves in Fig. 313 from data taken by F. C. Houghton and associates in Pittsburgh, show examples of seasonal variation in ground temperatures at different depths.

It would be difficult for the designer to calculate the required area of buried coils even if dependable information on soil conductivity were available because information is lacking on the proper thickness of soil to be considered between the coil surface and a point in the ground where the temperature is not appreciably affected by the heat transfer from or to the coil. Data on actual rates of heat transfer between buried coils of different diameter and spacing to or from the ground are sorely needed.

A test[4] was recently conducted with 40 ft of $\frac{3}{8}$-in. copper tubing buried to a depth of 5 ft and arranged in the form of a sinuous coil with the adjacent legs spaced 1.5 ft apart. Water was used as the circulating medium and entered the coil at 38 F. The equilibrium rate of heat transfer to this coil from soil whose temperature was 57 F was 600 Btu per hr or 6.24 Btu per hr per sq ft of pipe surface per deg F temperature difference between the soil and the circulating medium. This heat transfer was achieved with the coil buried in sandy soil containing only 0.038 lb of moisture per lb of dry sand and it is believed that this rate can safely be assumed for the estimation of required coil surface under practically all circumstances. When similar data are available for other soils it is believed that considerably higher rates can be used for clay soils or mixed soils and for all soils in localities where the annual rainfall is more than 20 in.

Prolonged tests covering a period of at least 2 years are needed to determine whether or not all the heat removed from the soil by the buried coils of a heat pump during a winter will be restored during the following summer.

PROBLEMS

1. The evaporator section of an air-conditioning unit serves as a water cooler through which 200 lb of water per min are passed to cool it from 56 F to 43 F.

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[4] "Heat Transfer from Ground to Water in Buried Coils," by William F. King and Michael G. Martus, Heating and Ventilating Magazine, p. 71, Vol. 44, July 1947.

The power input to the compressor is 15 bhp. Find the capacity of the unit in tons of refrigeration and its coefficient of performance.

2. A direct-expansion cooling coil serves as an evaporator and is operated with an air-vapor mixture leaving it at 60 F dbt and 58 F wbt when the barometric pressure is 28.5 in. of mercury. The air has an initial total enthalpy of 42.5 Btu per lb of dry air and the vapor which it contains. What tonnage of refrigeration capacity is developed when 12,000 lb of dry air per hr are handled? What is the coefficient of performance if 1.05 bhp are required per ton?

3. In the operation of a compression refrigeration system the working medium has the following enthalpies: leaving the compressor, 120; leaving the evaporator, 102; entering the evaporator, 22; and entering the compressor, 102. If 50 lb of the refrigerant are handled each minute what capacity, in tons, is being developed and what is the coefficient of performance?

4. A cooling tower may operate with an efficiency of 75 per cent. (*a*) Water at 90 F is sprayed into the air within the tower when the air is initially at 105 F dbt and 40 per cent relative humidity. What reduction of the water temperature is possible for the conditions stated? (*b*) What final water temperature may be expected if the outside-air temperature is at 90 F dbt and 75 F wbt?

CHAPTER 17

ESTIMATION OF COOLING LOADS

377. Foreword. The estimation of cooling loads on summer air-conditioning apparatus requires a more thorough analysis than is the case when calculating winter-heating requirements because items such as sun effect and heat liberated by occupants increase the peak load on the machine and, therefore, must be considered.

378. The Components of a Cooling Load. The two main divisions of a load imposed upon an air-conditioning plant operating during hot weather are: (1) the sensible heat to be removed, and (2) the water vapor to be abstracted.

Sensible heat to be removed in the process of summer air conditioning is derived from any or all the following sources.

1. Heat flowing into the building by conduction through walls, floors, ceilings, windows, skylights, and doors due to the difference between the outdoor-air temperature and the temperature of the air within the conditioned space.
2. Heat conducted through walls, roofs, windows, and doors to the interior of the building due to the absorption of solar radiation by certain of the exterior surfaces.
3. Heat given off by lights, motors, machinery, cooking operations, industrial processes, etc.
4. Heat liberated by the occupants.
5. Heat carried in by outside air which leaks in or is brought in for ventilation.
6. Heat gain through the walls of ducts carrying conditioned air through unconditioned spaces in the building which are at a temperature that is higher than that of the air handled.
7. Heat gain from fan work.

The water vapor which must be removed by the dehumidifying apparatus in order to maintain the specified relative humidity in the conditioned space may come from any or all the following sources.

1. Moisture in the outside air entering by infiltration or introduced for the purpose of ventilation.
2. Moisture from occupants.
3. Moisture from any process such as cooking which takes place within the conditioned space.

4. Moisture passing directly into the conditioned space through permeable walls or partitions from the outside or from adjoining regions where the water vapor pressure is higher.

In the process of removing water vapor by condensation the latent heat must be absorbed by the cooling apparatus, and the amount of this heat absorption required to condense the weight of water vapor which must be abstracted per hour is often referred to as the latent-heat load on the apparatus.

379. Outdoor Design Temperatures. The outdoor temperatures which should be assumed for the purpose of estimating cooling loads should not be the highest ever recorded, as such temperatures occur infrequently and only for periods of a few hours duration. In all large cities certain design dry-bulb and wet-bulb temperatures have become well established and may be used with confidence by engineers designing air-conditioning machines for use in that vicinity. Table 112 gives this information for many of the large cities of the country together with weather-bureau data on average wind velocities. Information in regard to the design wet-bulb temperature for a locality is

TABLE 112

DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER*

State	City	Design Dry-Bulb, Deg F	Design Wet-Bulb, Deg F	Summer Wind Velocity, Mph	Prevailing Summer Wind Direction
Ala.	Birmingham	95	78	5.2	S
	Mobile	92	79	8.6	SW
Ariz.	Phoenix	108	75	6.0	W
Ark.	Little Rock	96	79	6.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	92	64	6.9	S
Conn.	New Haven	95	75	7.3	S
D. C.	Washington	95	78	5.2	S
Fla.	Jacksonville	95	79	8.6	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	95	77	8.5	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	97	65	5.9	NW
Ill.	Chicago	95	75	10.0	NE
	Peoria	91	75	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	92	80	6.2	SW
Me.	Portland	90	73	7.3	S

TABLE 112 (*Continued*)

State	City	Design Dry-Bulb, Deg F	Design Wet-Bulb, Deg F	Summer Wind Velocity, Mph	Prevailing Summer Wind Direction
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	87	72	12.5	SW
Mich.	Detroit	93	73	10.2	SW
Minn.	Minneapolis	93	73	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	102	76	9.5	S
	St. Louis	100	76	9.4	SW
Mont.	Helena	89	62	8.0	SW
Neb.	Lincoln	95	75	9.3	S
Nev.	Reno	95	62	7.4	W
N. J.	Atlantic City	95	78	...	...
N. Y.	Albany	89	72	7.1	S
	Buffalo	87	72	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	90	78	6.9	SW
N. D.	Bismarck	95	73	8.8	NW
Ohio	Cleveland	93	73	11.2	S
	Cincinnati	95	77	5.6	SW
Okla.	Oklahoma City	100	76	10.1	S
Ore.	Portland	89	66	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	91	73	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	92	79	9.9	SW
	Columbia	95	78	...	...
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	98	79	7.4	SW
Tex.	Fort Worth	100	78	9.4	S
	Galveston	95	80	9.7	S
	San Antonio	98	77	7.3	SE
	Brownsville	92	79	7.0	S
	El Paso	97	69	8.6	E
Utah	Salt Lake City	97	63	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	80	65	8.1	S
	Spokane	93	62	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wis.	Green Bay	91	73	9.1	SW
	Milwaukee	95	75	10.4	S
Wyo.	Cheyenne	95	65	9.2	S

* Temperatures abstracted from "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

necessary as it is a reliable indication of the heat content of the outdoor air.

It may be noted from Table 112 that a design dry-bulb temperature of 95 F is used in many cities and that this figure is close to the temperature used in nearly all the cities for which data are given. Figure 315 shows a typical 24-hr plot of the outdoor dry-bulb temperature when the maximum temperature, occurring at 3:00 P.M., is 95 F.

380. Inside-Air Temperatures. The inside dry-bulb temperature maintained will of necessity be fixed either by the conditions necessary for comfort or by limitation of the outdoor-indoor-air temperature difference. A condition of 80 F and 50 per cent relative humidity has been found to be satisfactory for the majority of people and is often specified where human comfort is the only consideration. However, experience has shown that the average individual suffers an appreciable shock if subjected to a temperature change of more than 15 to 20 F as he enters or leaves a cooled space during hot weather. Therefore, in localities where the outside-air dry-bulb temperature often exceeds 100 F, the inside-air temperature should be maintained from 15 to 20 F below that of the outside air and an attempt made to produce comfort conditions by lowering the relative humidity of the conditioned space to some value below 45 per cent. The relative humidity desired at a given inside dry-bulb temperature will fix the wet-bulb temperature of the inside air and determine the amount of heat which must be removed from the outside air which infiltrates into the conditioned space or is purposely introduced for the purpose of ventilation. When it is not feasible to reduce the inside-air temperature below a certain limit because of a high dry-bulb temperature of

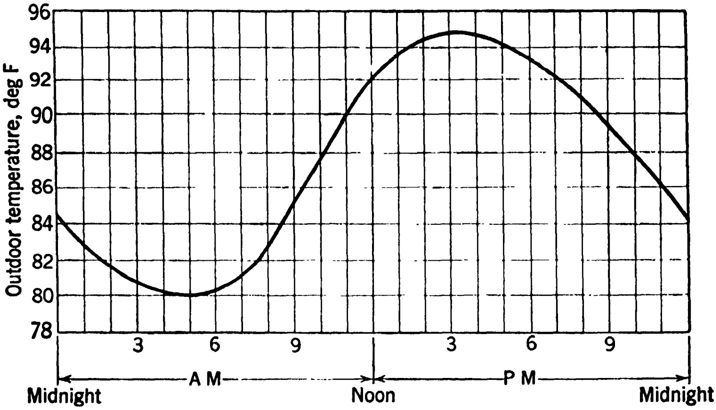


FIG. 315. Typical daily outdoor-air temperature variation during hot weather.

the outdoor air some additional lowering of the effective temperature of the inside air may be secured by slightly increased air motion. Greater outside-inside-temperature differences may be maintained in conditioned spaces such as homes in which the occupants remain for long periods than is the case with spaces such as department stores in which the occupancy of the customers is of brief duration. Residents of the northern part of the United States experience comfort conditions with lower inside-air temperatures than is the case with people living in the southern portion of the country. In the selection of the inside-air temperature to be maintained by a summer-cooling plant it is therefore necessary to consider the usage of the building and its geographical location.

381. Solar Radiation. Solar radiation striking the outside surfaces of a building may contribute appreciably to the peak load on summer air-conditioning apparatus and must therefore be considered. The amount of heat that flows toward the interior of a building owing to solar radiation depends on the altitude angle of the sun, the clearness of the sky, the position of the surface with respect to the direction of the sun's rays, the absorptivity of the surface, and the ratio of the over-all coefficient of heat transfer of the wall to the coefficient of heat transfer of the outside-air film. The altitude angle of the sun in turn depends upon the latitude of the locality, the season of the year, and the hour of the day.

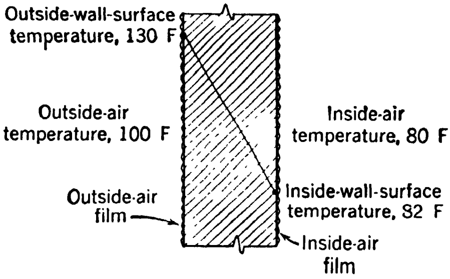

Outside-wall-surface temperature, 130 F
Outside-air temperature, 100 F
Outside-air film
Inside-air temperature, 80 F
Inside-wall-surface temperature, 82 F
Inside-air film

FIG. 316. Possible wall-surface temperatures produced by solar heat.

Heat from solar radiation is received by building surfaces in two forms, namely, direct radiation and sky radiation. Direct radiation is the impingement of the sun's rays upon the surface. Sky radiation is received from moisture and dust particles in the atmosphere which absorb part of the energy of the sun's rays, thereby becoming heated to a temperature above that of the air. Sky radiation is received by surfaces which do not face the sun.

Absorption of solar radiation by the outside surface of an opaque building wall or roof causes an increase in the temperature of the material at this surface. The outside-surface temperature usually becomes higher than that of the air on either side of the wall so that heat flows both ways from the heated surface (see Fig. 316). The flow of heat to the outdoor air and away from the building depends upon the temperature of the surface, the temperature of the outdoor air, and the coefficient of heat transfer for the outside air film; whereas the flow to the interior of the building depends upon the surface temperature, the inside-air temperature, and the overall coefficient of heat transfer between the heated material near the outside surface and the air inside the building.

A single thickness of transparent window glass transmits a large portion of any solar radiation striking it directly to the inside of the building where the energy becomes an instantaneous load on the cooling plant. However, thick walls or roofs may delay the interior effect of solar radiation as long as 16 hr. Table 113 gives the time lag in hours for several different roof and wall constructions. The time lag for a wall or roof construction not included in the table may be approximated by using the lag for a listed combination that is judged to be comparable with the one actually used. Because of time lag, the maximum cooling load from solar effect may occur several hours after the time of maximum solar intensity on the principal exposed building surfaces. For a building having very heavy masonry walls there may be no increase in the peak load on the cooling plant due to solar radiation because the wave of increased material temperature

TABLE 113

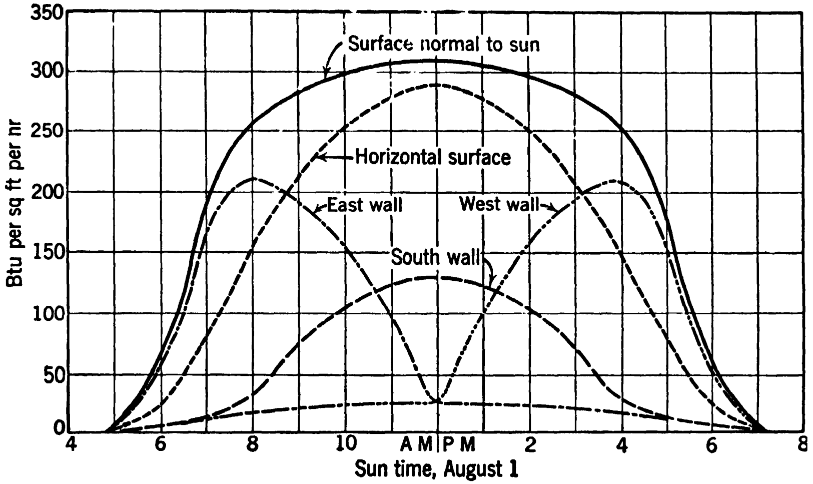
TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS*

Type and Thickness of Wall or Roof	Time Lag, Hr
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼-in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 by 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 by 4 studs (studding space filled with insulation, lath and plaster	5
4-in. brick, 1-in. sheathing, 2 by 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile, and 1¼-in. plaster	16

* From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

caused by the sun's rays may not reach the interior until after the load from all other sources has started to decline. On the other hand, time lag may cause a maximum cooling load to occur at an unexpected time as the wave of increased material temperature may reach the interior wall or roof surfaces at a time when the cooling load due to interior heat sources is a maximum as for a dance hall filled with people during the late evening

382. Sol-Air Temperature. A convenient way of combining all of the factors which affect the heat entry into the weather side of a sunlit building wall or roof is through the use of a term known as the sol-air temperature,[1] which is given by the following expression.



(*From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.*)

FIG. 317. Solar intensity at 40° north latitude on surfaces with different orientations.

$$t_s = T_o + \frac{aI}{f_o} \tag{144}$$

where t_s = sol-air temperature, deg F.

T_o = outside-air temperature, deg F.

a = portion of solar radiation I which is absorbed (see Table 114).

I = actual solar radiation striking surface (see Fig. 317), Btu per hr per sq ft.

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[1] "Summer Weather Data and Sol-Air Temperature—Study of Data for New York City," by C. O. Mackey and E. B. Watson, *Heating, Piping and Air Conditioning*, ASHVE Journal Section, November 1944.

f_o = outside-film coefficient of heat transfer (usually assumed to be 4.0 for summer conditions), Btu per hr per sq ft per deg F.

The sol-air temperature is the temperature of outdoor air which would give the same rate of heat entry into the weather side of a building material as exists with the actual combination of total radiation and outdoor-air temperature. For example, if $T_o = 95$, $a = 0.7$, $I = 150$, and $f_o = 4.0$, then $t_s = 95 + (0.7 \times 150) \div 4 = 25 + 26.25 = 121.25$ F.

TABLE 114

Solar Absorption Coefficients for Various Building Materials

Surface Color	Material	Coefficient a
Very light	White stone, very light-colored cement, white or light-cream-colored paint	0.4
Medium dark	Asbestos shingles, unpainted wood, brown stone, brick and red tile, dark colored cement, stucco, and red, green, or gray paint	0.7
Very dark	Slate roofing, tar roofing materials, and very dark paint	0.9

In terms of the sol-air temperature, the instantaneous rate of heat entry into the weather surface of a wall or roof is given by

$$H = Af_o(t_s - t_m) \tag{145}$$

where H = heat entering, Btu per hr.

A = roof or wall surface area, sq ft.

t_m = temperature of the wall or roof material at the weather surface, deg F.

Table 115 applicable to New York City gives summer design sol-air temperatures for each hour of the day. Data are included for horizontal roofs and for walls facing north, east, south, and west. The sol-air temperatures listed are for surfaces having an absorptivity of one or a value of the ratio $a/f_o = 0.25$. The table values must be corrected to the value of a/f_o that is applicable to the wall or roof in question. A similar table applicable to Lincoln, Nebraska, together with a further discussion of sol-air temperatures and a method of using them in the estimation of cooling loads is given in Chap. 15 of the "*Heating Ventilating Air Conditioning Guide* 1947." The sol-air temperature concept gives promise of providing a reasonably convenient and accurate method of estimating heat gain through walls and roofs when similar tables have been prepared for cities in other

TABLE 115

SUMMER DESIGN SOL-AIR TEMPERATURES FOR NEW YORK, N. Y.

North latitude 40° 46′; elevation 180 ft*

Sun Time	Sol-Air Temperature, Deg F					
	Any Surface	Horizontal	North	East	South	West
Ratio: $\frac{a}{f_o}$	0	0.25	0.25	0.25	0.25	0.25
12 midnight	89	89	89	89	89	89
1 A.M.	88	88	88	88	88	88
2	86	86	86	86	86	86
3	84	84	84	84	84	84
4	84	84	84	84	84	84
5	82	82	82	82	82	82
6	81	87	88	100	82	82
7	82	103	93	125	85	85
8	88	124	94	137	92	92
9	93	143	98	142	104	98
10	96	160	102	138	115	102
11	100	172	106	129	125	106
12 noon	102	178	108	115	130	108
1 P.M.	104	180	110	110	132	119
2	106	178	112	112	131	137
3	107	170	113	113	126	150
4	107	158	112	112	117	158
5	106	142	113	110	110	157
6	105	126	117	108	108	149
7	102	109	113	104	104	128
8	98	99	98	98	98	98
9	94	94	94	94	94	94
10	92	92	92	92	92	92
11	90	90	90	90	90	90
24-hr average t_m	94.4	121.6	98.6	105.9	102.1	106.6

* From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

parts of the country and the method has been proven reliable by more extensive experience with it.

383. Estimation of Sensible-heat Gain Through Glass Areas. Heat gain through single unshaded glass areas such as windows, doors, and skylights may be computed from values given in Table 116. The rates of heat gain which appear in this table include the heat that is conducted owing to temperature difference and that which is a result of total (direct and sky) solar radiation.

The figures which appear in this table have been computed for a solar declination angle (the angle between the direction of the sun's rays with respect to the earth and a plane perpendicular to the earth at the equator) of 18 deg. If it is desired to use the table for the

TABLE 116

TOTAL INSTANTANEOUS RATES OF HEAT GAIN THROUGH SINGLE, UNSHADED, COMMON GLASS, FOR VARIOUSLY ORIENTED VERTICAL AND FOR HORIZONTAL POSITIONS*

Data for solar declination of 18 deg—August 1

Sun Time	Solar Altitude, β Deg	Total Instantaneous Rate of Heat Gain, Btu per Hr for Each Sq Ft of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
					25 Deg North Latitude					
6 A.M.	7.5	19	77	84	40	3	3	3	3	12
7	20.5	26	146	173	98	10	10	10	10	68
8	34.0	18	148	193	121	13	13	13	13	143
9	47.5	15	118	172	120	16	15	15	15	205
10	61.5	16	67	124	95	22	16	16	16	252
11	74.5	16	25	57	57	25	16	16	16	282
12	83.0	16	16	16	25	28	25	16	16	293
1 P.M.	74.5	16	16	16	16	25	57	57	25	282
2	61.5	16	16	16	16	22	95	124	67	252
3	47.5	15	15	15	15	16	120	172	118	205
4	34.0	18	13	13	13	13	121	193	148	143
5	20.5	26	10	10	10	10	98	173	146	68
6	7.5	19	3	3	3	3	40	84	77	12
					30 Deg North Latitude					
6 A.M.	9.0	22	88	97	47	4	4	4	4	15
7	21.5	23	146	176	105	11	11	11	11	74
8	34.5	16	140	194	130	14	14	14	14	144
9	47.5	15	104	171	133	20	15	15	15	205
10	60.0	16	53	126	112	33	16	16	16	248
11	72.0	16	19	56	74	42	16	16	16	277
12	78.0	16	16	16	34	45	34	16	16	288
1 P.M.	72.0	16	16	16	16	42	74	56	19	277
2	60.0	16	16	16	16	33	112	126	53	248
3	47.5	15	15	15	15	20	133	171	104	205
4	34.5	16	14	14	14	14	130	194	140	144
5	21.5	23	11	11	11	11	105	176	146	74
6	9.0	22	4	4	4	4	47	97	88	15
					35 Deg North Latitude					
6 A.M.	10.0	21	97	109	53	4	4	4	4	19
7	22.5	19	143	179	110	11	11	11	11	79
8	34.5	14	133	194	140	15	14	14	14	144
9	46.5	15	90	170	144	28	15	15	15	200
10	58.5	16	38	126	128	47	16	16	16	243
11	68.5	16	16	56	91	62	17	16	16	269
12	73.0	16	16	16	45	68	45	16	16	279
1 P.M.	68.5	16	16	16	17	62	91	56	16	269
2	58.5	16	16	16	16	47	128	126	38	243
3	46.5	15	15	15	15	28	144	170	90	200
4	34.5	14	14	14	14	15	140	194	133	144
5	22.5	19	11	11	11	11	110	179	143	79
6	10.0	21	4	4	4	4	53	109	97	19

TABLE 116 (*Continued*)

Sun Time	Solar Altitude, β Deg	Total Instantaneous Rate of Heat Gain, Btu per Hr for Each Sq Ft of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
						40 Deg North Latitude				
5 A.M.	1.5	7	18	17	6	1	1	1	1	2
6	11.5	23	106	120	62	5	5	5	5	24
7	23.0	15	141	181	118	11	11	11	11	82
8	34.5	14	122	194	147	19	14	14	14	145
9	45.5	15	76	172	156	42	15	15	15	196
10	56.0	16	30	125	144	66	16	16	16	235
11	64.5	16	16	53	110	85	22	16	16	261
12	68.0	16	16	16	62	94	62	16	16	269
1 P.M.	64.5	16	16	16	22	85	110	52	16	261
2	56.0	16	16	16	16	66	144	125	30	235
3	45.5	15	15	15	15	42	156	172	76	196
4	34.5	14	14	14	14	19	147	194	122	145
5	23.0	15	11	11	11	11	118	181	141	82
6	11.5	23	5	5	5	5	62	120	106	24
7	1.5	7	1	1	1	1	6	17	18	2
						45 Deg North Latitude				
5 A.M.	2.0	9	23	23	8	1	1	1	1	2
6	12.5	22	111	129	68	6	6	6	6	28
7	23.0	13	135	182	121	11	11	11	11	82
8	33.5	14	116	192	153	24	14	14	14	141
9	44.0	15	63	168	166	55	15	15	15	189
10	53.0	16	22	123	154	88	16	16	16	225
11	60.0	16	16	56	127	113	30	16	16	249
12	63.0	16	16	16	76	119	76	16	16	256
1 P.M.	60.0	16	16	16	30	113	127	56	16	249
2	53.0	16	16	16	16	88	154	123	22	225
3	44.0	15	15	15	15	55	166	168	63	189
4	33.5	14	14	14	14	24	153	192	116	141
5	23.0	13	11	11	11	11	121	182	135	82
6	12.5	22	6	6	6	6	68	129	111	28
7	2.0	9	1	1	1	1	8	23	23	2
						50 Deg North Latitude				
5 A.M.	4.5	18	48	48	17	2	2	2	2	5
6	13.5	22	119	139	75	6	6	6	6	32
7	23.5	11	131	183	127	11	11	11	11	84
8	33.0	13	103	190	161	30	13	13	13	138
9	42.0	14	51	165	173	69	14	14	14	179
10	50.0	16	19	122	166	109	16	16	16	214
11	56.0	16	16	55	138	133	41	16	16	235
12	58.0	16	16	16	92	140	92	16	16	242
1 P.M.	56.0	16	16	16	41	133	138	55	16	235
2	50.0	16	16	16	16	109	166	122	19	214
3	42.0	14	14	14	14	69	173	165	51	179
4	33.0	13	13	13	13	30	161	190	103	138
5	23.5	11	11	11	11	11	127	183	131	84
6	13.5	22	6	6	6	6	75	139	119	32
7	4.5	18	2	2	2	2	17	48	48	5

* From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

determination of heat gain at a time of the year when the solar declination angle is greater or less than 18°, use the latitude which differs from that of the locality by the difference between the actual solar-declination angle and 18°. For example, if it is desired to compute heat gain for a time when the solar declination toward the north is 23°, the section of the table for a latitude 5° greater than the actual latitude would be used. Table 117 gives solar declination angles for the first and fifteenth of each month during the year 1947. These data are strictly applicable only to the year for which they were computed, but the declination angle for a given day of the year varies only slightly from year to year.

All the data in Table 116 are for common glass having a transmissivity of 0.87. Certain special heat-absorbing glasses retain more of the solar radiation and transmit less of it to the inside of the building because, with this type of glass, the temperature of the pane rises to a level that is well above the outdoor-air temperature. As a result, an appreciable portion of the heat entering the weather side of the glass due to solar radiation is conducted back to the outdoors through the outside-air film and carried away by the outside air.

TABLE 117

APPARENT DECLINATION OF THE SUN FOR DIFFERENT TIMES OF THE YEAR*

Month	Day	Declination Angle	Month	Day	Declination Angle
January	1	S 23° 03′	July	1	N 23° 09′
January	15	S 21° 14′	July	15	N 21° 37′
February	1	S 17° 15′	August	1	N 18° 10′
February	15	S 12° 52′	August	15	N 14° 15′
March	1	S 7° 48′	September	1	N 8° 30′
March	15	S 2° 22′	September	15	N 3° 15′
April	1	N 4° 19′	October	1	S 2° 58′
April	15	N 9° 34′	October	15	S 8° 18′
May	1	N 14° 54′	November	1	S 14° 15′
May	15	N 18° 44′	November	15	S 18° 21′
June	1	N 21° 59′	December	1	S 21° 43′
June	15	N 23° 17′	December	15	S 23° 15′
June	22 (max)	N 23° 27′	December	22 (max)	S 23° 27′

* Abstracted from the "American Air Almanac, 1947." Declination angles given are for noon of each day listed.

Table 118 gives transmissivity ratios for special glasses for which the transmissivity is less than that for common glass. These ratios are to be used in correcting the values for heat gains from Table 116 when estimating heat gains through special glasses. Transmissivities of special glasses are furnished upon request by the manufacturers.

TABLE 118

RATIO OF TOTAL INSTANTANEOUS RATE OF HEAT GAIN THROUGH SINGLE THICKNESS OF SUNLIT GLASS TO HEAT GAIN FOR COMMON GLASS AS GIVEN IN TABLE 116*

Transmissivity	Approximate Ratio	Transmissivity	Approximate Ratio	Transmissivity	Approximate Ratio
0.87	1.0	0.60	0.76	0.30	0.52
0.80	0.93	0.50	0.68	0.20	0.45
0.70	0.85	0.40	0.60		

* From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

Heat gains through glass of any sort can be greatly reduced by shading. The fraction of the heat transmitted by unshaded glass that would be transmitted by glass that is shaded in different ways is given in Table 119.

TABLE 119

EFFECT OF SHADING ON TOTAL RATES OF INSTANTANEOUS HEAT GAIN THROUGH GLASS*

Type of Shading	Finish	Fraction of Gain through Unshaded Window
Outside shading screen, metal slats 0.05 in. wide, spaced 0.063 in. apart, and set at 17° angle with horizontal	Dark	0.20 to 0.35
Canvas awning	Dark	0.25 to 0.35
Outside venetian blind, slats at 45° extended as an awning without sides, to cover approximately two-thirds of window	Light	0.35 to 0.50
Inside roller shade, fully drawn	Aluminum	Approximately 0.45
Outside venetian blind, slats at 45° fully covering window	Aluminum	Approximately 0.3
Inside venetian blind, slats at 45° fully covering window	Aluminum	0.65 to 0.80
Inside roller shade, half drawn	Buff	Approximately 0.70
Inside roller shade, half drawn	Dark	0.90 to 0.95

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Windows and doors which are set in from the plane of the weather side of a vertical wall will have a portion of their areas shaded by the projecting edges of the openings. If a vertical window or door (see Fig. 318) of height h and width w is set back from the plane of the weather side of the wall a distance d, the fraction of the total area which receives direct solar radiation is given by the following equation

$$G_f = 1 - r_1 \tan \beta - r_2 \tan \gamma + r_1 r_2 \tan \beta \tan \gamma \qquad (146)$$

where $r_1 = d/h$.

$r_2 = d/w$.

β = the solar altitude angle.

γ = the angle between the sun's rays and the vertical faces of the window opening.

Solar altitude angles for August 1 are given in Table 116, and Table 120 includes values for the angle γ.

Similar tables can be prepared for other dates if desired. The effect of the shading of glass areas by the walls in which they are set can usually be neglected, but it may appreciably affect the specified capacity of the cooling apparatus when the major heat gain is through a wall containing a considerable proportion of windows or doors which are set back from the face of the wall.

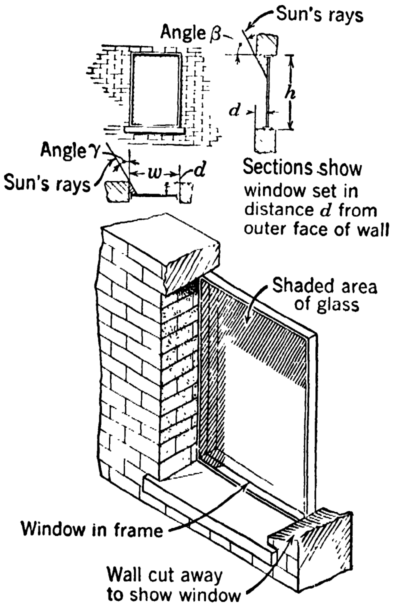

FIG. 318. Shading of the upper portion and one edge of either a window or a door with frame inset from the weather surface of a wall.

Example. Estimate the instantaneous heat gain at 3 P.M. on August 1 through a west window of common glass measuring 4 ft wide by 8 ft high and set back 9.6 in. from the face of the wall. The building is situated in a locality that is 40 deg north latitude.

Solution. $r_1 = 9.6 \div 96 = 0.1$, and $r_2 = 9.6 \div 48 = 0.2$. The altitude angle, from Table 116, is 45.5° and its tangent equals 1.02. Table 120 indicates angle γ to be 16° whose tangent is 0.267. Therefore, from equation 146, $G_f = 1 - 0.1 \times 1.02 - 0.2 \times 0.267 + 0.1 \times 0.2 \times 1.02 \times 0.267 = 1 - 0.102 - 0.0534 + 0.0054 = 0.8392$. Thus approximately 84 per cent of the area of this window will receive direct solar radiation at the time stated. From Table 116 it is found that the heat gain per square foot of unshaded area is 42 Btu per hr. Although the heat conducted due to temperature difference and the heat gain due to sky radiation is not reduced by the shading caused by a projecting ledge it is approximately correct to estimate the heat gain of this window as $H = 4 \times 8 \times 0.84 \times 42 = 1130$ Btu per hr. If shading due to projecting ledges were not considered, the heat gain of this window would be computed as $4 \times 8 \times 42 = 1344$ Btu per hr.

384. Heat Conduction through Multiple-Glass Layers and Glass Blocks. When two or more glass layers separated by air spaces are

TABLE 120

VALUES OF THE AZIMUTH-DIFFERENCE ANGLE, γ*, DEGREES, FOR WINDOW-REVEAL SHADING CALCULATIONS (SEE EQUATION 146)

Computed for solar declination of 18 deg—August 1

North Latitude, Deg	Mean Sun Time	Window Orientation						
		NE	E	SE	S	SW	W	NW
30	6 A. M.	29	16	61	Shade	Shade	Shade	Shade
	7	36	9	54	. .	. .	. .	. .
	8	43	2	47	. .	. .	. .	. .
	9	51	6	39	84	. .	. .	. .
	10	62	17	28	73	. .	. .	. .
	11	83	38	7	52	. .	. .	. .
	12 noon	Shade	Shade	45	0	45	. .	. .
	1 P. M.	. .	. .	Shade	52	7	38	93
	2	. .	. .	. .	73	28	17	62
	3	. .	. .	. .	84	39	6	51
	4	. .	. .	. .	Shade	47	2	43
	5	. .	. .	. .	. .	54	9	36
	6	. .	. .	. .	. .	61	16	29
40	6 A. M.	31	14	59	Shade	Shade	Shade	Shade
	7	40	5	50	. .	. .	. .	. .
	8	50	5	40	85	. .	. .	. .
	9	63	16	29	74	. .	. .	. .
	10	76	31	14	59	. .	. .	.
	11	Shade	55	10	35	. .	. .	. .
	12 noon	. .	Shade	45	0	45	. .	. .
	1 P. M.	. .	. .	80	35	10	55	. .
	2	. .	. .	Shade	59	14	31	76
	3	. .	. .	. .	74	29	16	63
	4	. .	. .	. .	85	40	5	50
	5	. .	. .	. .	Shade	50	5	40
	6	. .	. .	. .	. .	59	14	31
50	6 A. M.	33	12	57	Shade	Shade	Shade	Shade
	7	45	0	45	. .	. .	. .	. .
	8	57	12	33	78	. .	. .	. .
	9	70	25	20	65	. .	. .	. .
	10	87	42	3	48	. .	. .	. .
	11	Shade	64	19	25	71	. .	. .
	12 noon	. .	Shade	45	0	45	. .	. .
	1 P. M.	. .	. .	71	25	19	64	. .
	2	. .	. .	Shade	48	3	42	87
	3	. .	. .	. .	65	20	25	70
	4	. .	. .	. .	78	33	12	57
	5	. .	. .	. .	Shade	45	0	45
	6	. .	. .	. .	. .	57	12	33

---

* γ is the angle between the horizontal projection of the sun's rays and the plane of either vertical face of the window or door opening.

used instead of a single pane, the absorptivity is increased, and the transmissivity is decreased. Data on heat gain through all such arrangements are not available, but a rough approximation may be made of the heat transmitted by assuming that the transmissivity of the combination is the product of the transmissivities of the component layers. Thus the transmissivity of two layers of common glass could be assumed to be 0.87 × 0.87 = 0.757. After estimating the transmissivity of a glass area consisting of two or more layers, the proper correction factor to be applied to the values of heat gain from Table 116 may be determined by interpolation from Table 118.

For example, if a window consisted of three layers of special glass, each having a transmissivity of 0.7, the transmissivity of the combination would be assumed to be 0.7 × 0.7 × 0.7 = 0.343. Interpolating in Table 118 the ratio of the heat gain through this combination to that through a single thickness of common glass is found to be approximately 0.56. The heat gain through a glass area of this type would

TABLE 121

INSTANTANEOUS TOTAL RATES OF HEAT GAIN THROUGH GLASS BLOCKS ON AUGUST 1*

Sun Time	Total Instantaneous Rate of Heat Gain, Btu per Hr for Each Sq Ft of Sunlit Glass					
	Vertical Surface Facing					
	East	West	South			
North Latitude, Deg	30 to 45	30 to 45	30	35	40	45
7 A.M.	61	. .	−4.5	−2.0	−0.5	1.0
8	78	. . .	0.0	2.0	4.0	5.0
9	74	5.0	5.0	7.0	10	12
10	58	6.5	11	15	18	21
11	45	7.5	17	22	26	32
12 noon	37	11	22	28	34	41
1 P.M.	30	22	25	32	39	46
2	24	35	26	32	39	47
3	20	55	24	30	37	45
4	16	77	20	26	32	41
5	13	86	15	20	25	34
6	11	55	9.5	14	18	26
7	8	19	3.5	7.0	11	18

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then be assumed as 56 per cent of that computed for an equal area of common glass using the data in Table 116.

The instantaneous total rates of heat gain through glass blocks is given in Table 121. The table is based on a temperature of the indoor air of 78 F, and the values given are for a day having a maximum outdoor dry-bulb temperature of 95 F. The values given were obtained at the ASHVE Research Laboratory and are average for four typical glass-block designs, two with smooth exterior faces and two with ribbed exteriors.

385. Heat Gain through Outside Walls and Roofs. Heat gain through building walls and roofs which are not exposed to the sun's rays is given under conditions of steady flow by equation 147 and may be calculated in exactly the same way that heat losses are estimated.

$$H = UA(t_o - t_i) \tag{147}$$

where H = heat transmitted, Btu per hr.
U = overall coefficient of heat transfer, Btu per hr per sq ft per deg F.
A = roof or wall area, sq ft.
t_o = outside dry-bulb design temperature, deg F.
t_i = inside dry-bulb design temperature, deg F.

Although this expression does not give the exact amount of heat delivered to the interior of a building when the outside temperature is varying because of time lag, it may be used without introduction of serious error because the temperature of the weather surface of such a wall or roof never exceeds the outdoor design dry-bulb temperature. However, for a roof or wall which is exposed to the sun's rays, the temperature of the weather side may exceed the design outdoor-air temperature by many degrees, and it is necessary to consider time lag for reasons which have been explained in Art. 381.

All four walls of a building standing by itself will receive direct radiation from the sun at certain times each day during the summer. However, as has been mentioned previously, absorption of solar energy by the weather side of a building wall or roof does not result in an immediate heat gain in the interior. Heat which may be expected to actually flow through to the inside air through either roofs or walls when in any of the common orientations are given for all hours of the day and for several different types of construction by the curves shown in Figs. 319 to 324 inclusive. Figure 324 which is applicable to west walls can also be used for walls facing southwest, and Fig. 322 for east walls can be applied to walls facing southeast. Figures

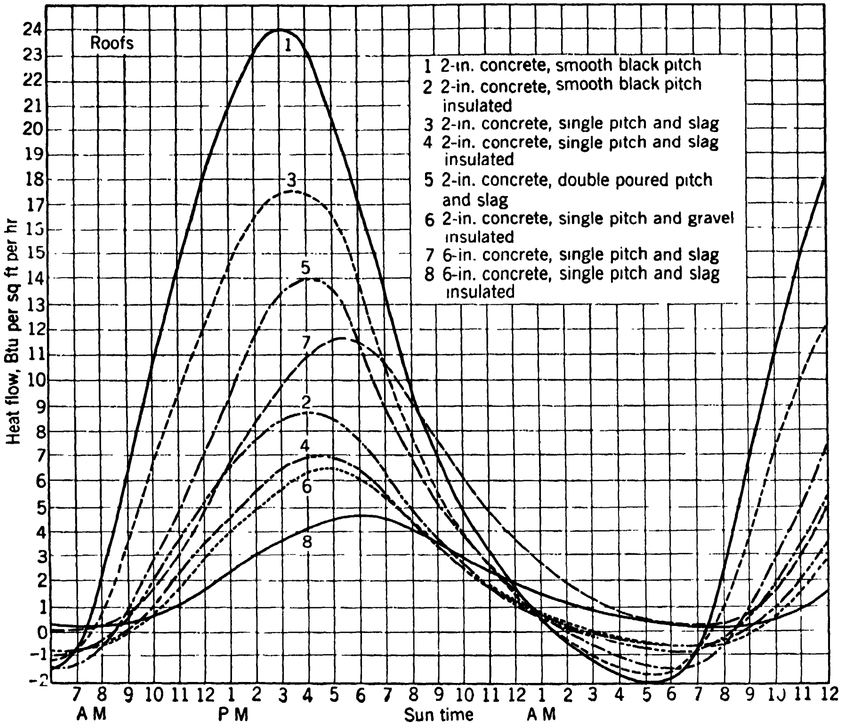

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FIG. 319. Heat-flow-time relationship for horizontal roofs.

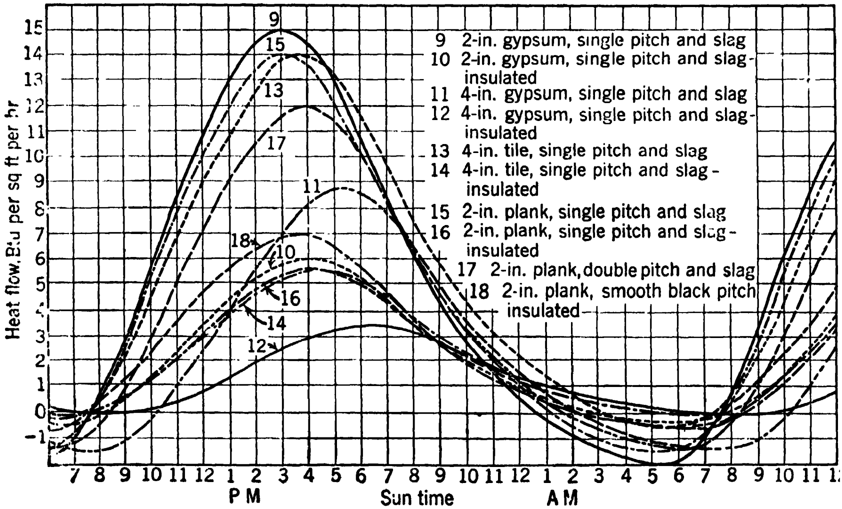

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FIG. 320. Heat-flow-time relationship for horizontal roofs.

319 to 324 inclusive were derived from the data of Fig. 317 taking into consideration the time lag for each wall or roof construction for which a curve was prepared.

All the data which are presented in graphical form in Fig. 317 and in Figs. 319 to 324 inclusive were accumulated at the ASHVE Research

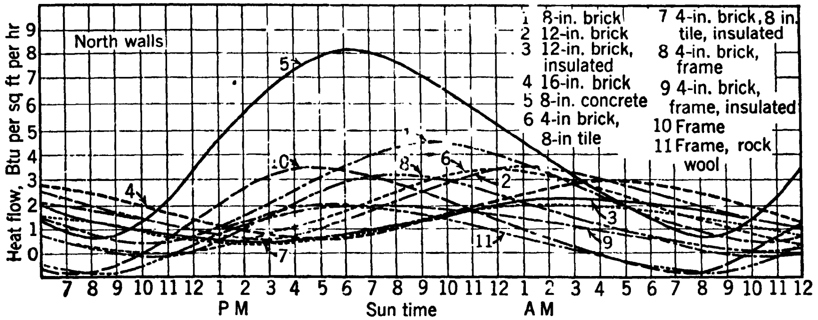

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FIG. 321. Heat-flow-time relationship for northern exposed walls.

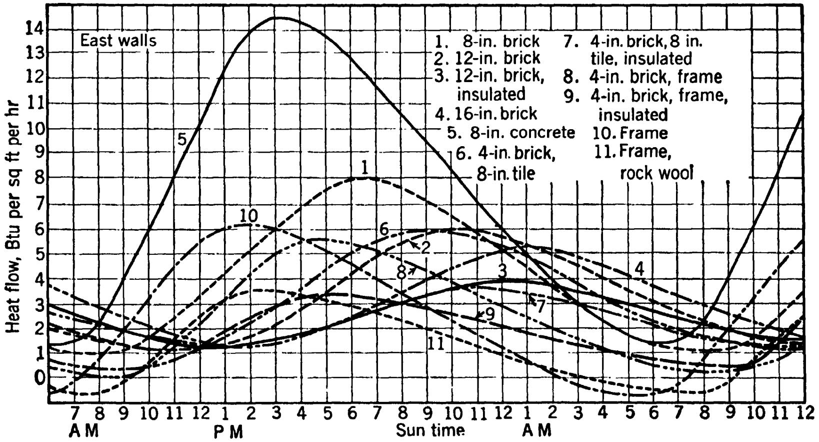

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FIG. 322. Heat-flow-time relationship for eastern exposed walls.

Laboratory at Pittsburgh, Pennsylvania, where the atmosphere is somewhat hazy the greater part of the time. Many engineers believe they should be increased as much as 30 per cent when used for estimating the load on summer cooling apparatus which is to serve a building in a locality where the atmosphere is comparatively clear. Likewise these data are based on a maximum outdoor dry-bulb temperature of 95 F occurring at 3 P.M. and should be increased propor-

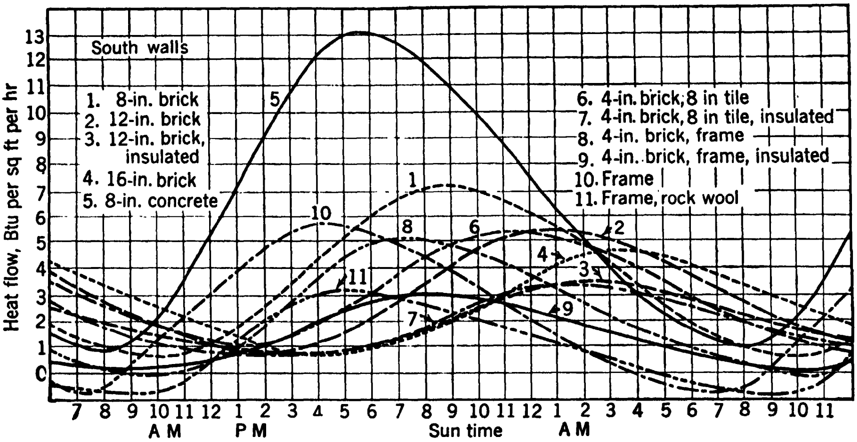

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FIG. 323. Heat-flow-time relationship for southern exposed walls.

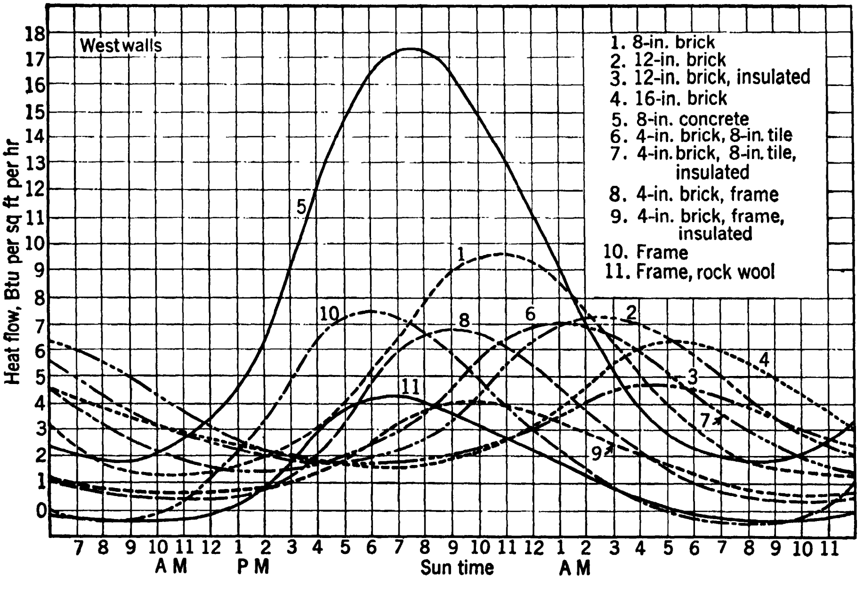

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FIG. 324. Heat-flow-time relationship for western exposed walls.

tionately if used in localities where the proper outdoor design temperature is greater than this figure.

The latitude of a locality has little effect on the maximum rate of heat gain by the weather surface of vertical walls facing either east or west. Therefore, in estimating the cooling load for a building in which the gain through either of these walls is the major part of the maximum cooling load, the proper curve in either Fig. 322 or 324 could be used with slight error regardless of the geographical location of the building. However, the maximum amount of heat entering the weather side of a south wall would vary with the latitude of the location approximately the same as the variation in the heat gain through common glass as given in Table 116. Until more complete information is available it is recommended that the following procedure be used for estimating the heat gain through a south wall when the building is located in a locality that is a considerable distance north or south of 40° north latitude for which the curves of Figs. 319 to 324 were prepared.

1. Determine the heat gain through the south wall in question for the time of day when the maximum cooling load is expected to occur, using the proper curve in Fig. 323.

2. Multiply the heat gain of the south wall as estimated in step 1 by the ratio of the heat entering the weather surface of a wall in the actual latitude to the heat entering the same surface of a comparable wall located at 40° north latitude. This ratio may be assumed to equal the ratio of the heat gains through common glass at the two different latitudes.

The proposed method can best be illustrated by an example as follows:

Example. Find the heat gain at 4 P.M. on August 1 through one square foot of a south wall consisting of wood siding, one-inch sheathing, 2 by 4 in. studs, lath and plaster, located at 30 deg north latitude.

Solution. From curve 10, Fig. 323, the heat gain through a frame wall at 4 P.M. is found to be 5.7 Btu per sq ft per hr. From Table 113 it may be noted that the time lag for this type of wall is 2 hr so that the heat flowing through the inside surface at 4 P.M. is determined by the heat entering the outside surface at 2 P.M. The heat entering the outside surface may be assumed to be proportional to the heat transmitted through common glass as given in Table 116. The heat transmitted through one square foot of common glass in a south wall at 2 P.M. in 30° north latitude is 33 Btu per hr and in 40° north latitude is 66 Btu per hr. The rate of heat entry into the south wall at this hour is then only one-half what it would have been if the building had been located in the latitude for which the curves were drawn. The rate of heat gain through the wall of the example at 4:00 P.M. may then be assumed to be $(33 \times 5.7)/66 = 2.85$ Btu per hr per sq ft.

Examination of Table 116 discloses the fact that the heat gain through a single pane of common glass for any particular hour through the warmest part of the day varies little with the latitude of the locality in regard to east or west exposures. Since the variation in the declination angle of the sun due to the progress of the season affects the direction of the sun's rays with respect to the east and west walls of a building in the same manner as variation in the latitude of the locality for a given date such as August 1, it may be observed that the time of the year has little effect on the amount of solar radiation striking an east or a west wall during the warmest part of each day. Therefore the curves in Figs. 322 and 324 which were prepared for August 1 could be used for an earlier or a later date. A rough approximation of the heat gain through a south wall for a date other than August 1 can be made by using the method illustrated for making a correction for latitude. In making such an approximation use the section of Table 116 which applies to the latitude obtained by subtracting the difference between the actual declination angle of the sun, Table 117, and 18 from the degree latitude for the location of the building under consideration.

Example. Find the heat gain through one square foot of the south wall in the preceding example for July 2 instead of for August 1.

Solution. From Table 117 it is found that the declination of the sun on July 2 is approximately 23. Therefore the corrected latitude is $30 - (23 - 18) = 30 - 5 = 25$ deg north, and the section in Table 116 applying to 25° north latitude should be used. The heat gain through common glass at 2 P.M. when located at 25° north latitude is found to be 22 Btu per hr instead of 66 Btu per hr when at 40 deg north latitude. The rate of heat gain through one square foot of this wall on July 2 would then be estimated as $(22 \times 5.7)/66 = 1.9$ Btu per hr.

386. Conduction Heat Gains through Interior Partitions, Floors or Ceilings. It seldom happens that the entire interior volume of a building is air conditioned, so it is necessary to consider the heat that may be gained through either vertical or horizontal partitions separating the conditioned spaces from other portions of the building. Heat gain from this source may be computed by application of the following equation.

$$H = UA(t_n - t_c) \tag{148}$$

where H = sensible heat gain, Btu per hr.

U = overall coefficient of heat transfer, Btu per hr per sq ft per deg F temperature difference.

A = area of the partition, sq ft.

t_n = temperature in the adjacent space which is not conditioned, deg F.

t_c = temperature in the conditioned space, deg F.

U may be calculated by means of either equation 44 or equation 45, Chap. 4, or obtained from a suitable table.

In every case it will be necessary for the designer to estimate the temperature in the unconditioned space on the opposite side of each interior partition. The temperature in a well-ventilated room above ground level may be assumed to be the same as the temperature of the outdoor air. Attic temperatures will depend principally upon the amount of ventilation provided but will usually be higher than those in other unconditioned spaces. The air temperature in an attic that is adequately provided with cross ventilation may be estimated as 5 to 10 F above the outdoor temperature. However, when no ventilation is provided, an attic temperature may reach a value up to 35 deg above that of the outdoor air. Heat gain through first-story floors over basements can usually be neglected and sometimes may be negative as the basement temperature may be lower than that of a conditioned space above it. If considerable heat is liberated in an unconditioned room by processes such as either washing clothes or cooking food the temperature in this space may be assumed as 15 deg above that of the outside air under summer design conditions.

387. Heat and Moisture Emission of Appliances. Appliances frequently used in air-conditioned spaces which may liberate heat or water vapor may be electrical, gas fired, or steam heated. An analysis must be made of the operation of all such appliances and the heat and moisture emissions to the conditioned space determined. Table 122 gives a tabulation of several of the commonly used appliances together with approximate values for the sensible heat and moisture emitted when they are used. The increase in the latent-heat load on the dehumidifying apparatus due to the emission of moisture by the appliances is also given.

For a concentrated source of heat or moisture or both such as a steam table in a restaurant, it may be advisable to install a hood over the heated area to collect the warm air or water vapor rising from it, thus preventing it from becoming an added load on the cooling and dehumidifying equipment. However, to be effective, hoods must be connected by a duct to an exhaust fan capable of moving sufficient air to create a face velocity of at least 50 fpm. Even well-designed hoods over steam tables or similar appliances cannot be counted upon for the removal of more than 50 per cent of the heat emitted by them.

Table 123 gives recommended allowances for heat gain from electric motors in different positions with respect to the conditioned space.

388. Heat and Moisture Liberated by People Occupying a Conditioned Space. When a conditioned space is occupied by a large num-

TABLE 122

ALLOWANCES TO BE MADE IN THE ESTIMATION OF COOLING LOADS FOR THE HEAT AND MOISTURE EMISSION OF APPLIANCES

Appliance	Sensible Heat Emitted, Btu per Hr	Moisture Emitted, Grains per Hr	Increase in Latent-Heat Load, Btu per Hr
Electric lights	Total Wattage Times 3.416	0	0
Electric baking ovens, toasters, food warming receptacles, etc.	Total Wattage Times 2.733, Assuming 80 Per Cent of Total	Total Wattage Times 4.55	Total Wattage Times 0.683, Assuming 20 Per Cent of Total
Heat liberated from hot water direct and on towels in a barber shop	100	1335	200
Hair dryer, blower type	2300	2670	400
Hair dryer, helmet type	1870	2200	330
Permanent-wave machine	850	1000	150
Instrument sterilizer	650	8000	1200
Small bunsen burner using manufactured gas	960	1600	240
Small bunsen burner using natural gas	1680	2800	420
Fish-tail burner using manufactured gas	1960	3260	490
Fish-tail burner using natural gas	3080	5130	770
Natural gas burned in unvented burner, per cu ft of gas burned per hr	900	667	100
Manufactured gas burned in unvented burner, per cu ft of gas burned	540	400	60
Producer gas burned in unvented burner, per cu ft of gas burned	135	100	15
Continuous-flame cigar lighter	900	667	100
Coffee urn, per gal of capacity	1025	6830	1025
Heat liberated by food in a restaurant, per person	30	200	30
Steam table, per sq ft of top surface	300	5330	800
Steam heated surface, not polished, per sq ft	330	0	0
Steam heated surface, polished, per sq ft	130	0	0
Bare steam pipes, per sq ft of surface	400	0	0
Insulated steam pipes, per sq ft of surface	110	0	0

TABLE 123

SENSIBLE HEAT GAIN FROM ELECTRIC MOTORS

Name-Plate Rating, Hp	Heat Gain, Btu per Hr per Hp of Rating: Connected Load in Air-Conditioned Space, Motor Outside	Motor in Air-Conditioned Space, Load Outside	Motor and Connected Load Both in Air-Conditioned Space
$\frac{1}{20}$ to $\frac{1}{8}$	2546	2354	4900
$\frac{1}{8}$ to $\frac{1}{2}$	2546	1 74	3920
$\frac{1}{2}$ to 3	2546	724	3220
3 to 20	2546	454	3000
20 to 150	2546	284	2830

ber of people the heat and moisture emitted from their bodies may constitute the major portion of a summer cooling load. The heat and moisture emission from the human body under different circumstances has been discussed in Chap. 3. Table 124 is provided for the convenience of the reader.

TABLE 124

HEAT AND MOISTURE EMITTED BY OCCUPANTS OF AN AIR-CONDITIONED ROOM MAINTAINED AT 79 F*

Description of Occupants' Activity	Total Heat, Btu per Hr	Sensible Heat, Btu per Hr	Latent Heat, Btu per Hr	Moisture, Grains per Hr
Seated and at rest	384	225	159	1072
Seated but moderately active (office worker)	490	225	265	1787
Standing but not exercising (reception)	431	225	206	1389
Moderately active (store clerk)	600	225	375	2529
Continuous walking at 2 mph or light dancing	761	250	511	3446
Rapid walking at 4 mph, active dancing or roller skating	1390	452	938	6325
Light metal worker at bench	862	277	585	3945
Bowling	1500	490	1010	6811
Men sawing wood (heavy labor)	1800	590	1210	8160

* Abstracted by permission from a table prepared by the Air Conditioning Department, General Electric Company.

389. Heat and Moisture Brought into Conditioned Spaces by Outdoor Air. Since the temperature of the outdoor air is greater than that of the spaces served, under design conditions, the sensible-heat load on the cooling apparatus is increased in an amount that is directly proportional to the amount of this air that finds its way into the conditioned rooms. Outdoor air may be brought in deliberately for the purpose of providing ventilation, or it may come in by infiltration through cracks in the building structure. In addition to increasing the sensible heat load, the introduction of outdoor air usually brings in excess moisture which increases the relative humidity in the conditioned rooms unless it is removed by condensation in the equipment that is provided for dehumidification. The correct design of a summer air-conditioning machine requires an estimation of both the sensible heat gain and either the latent heat gain or the moisture gain due to the introduction of outdoor air into the space that is to be conditioned. The following expressions may be used.

$$H_t = W(h_o - h_i) \tag{149}$$

$$M = \frac{W(m_o - m_i)}{7000} \tag{150}$$

$$H_L = M \times h_{fg} \tag{151}$$

$$H_s = H_t - H_L \tag{152}$$

where H_t = total heat gain due to introduction of outdoor air, Btu per hr.

W = estimated weight of outdoor air entering conditioned space, lb of dry air per hr.

h_o and h_i = enthalpies of outdoor and inside air respectively, Btu per lb of dry air.

M = moisture gain, lb per hr.

m_o and m_i = humidity ratio of outdoor and inside air respectively, grains per lb of dry air.

H_L = latent-heat gain, Btu per hr.

h_{fg} = latent heat of vaporization of water vapor, average value 1050 Btu per lb.

H_s = sensible-heat gain, Btu per hr.

Ventilation requirements are discussed in Chap. 14. Infiltration may be estimated by the method that is given in Chap. 4 except that the density used in converting volume to weight would be lower owing to the higher temperature of the air. In general the weight of outdoor air entering a building by infiltration is less in summer than in cold weather because of lower air density, lower wind velocity, and a smaller difference between the densities of the outdoor and indoor air. Infiltration of outdoor air may be estimated by assuming a reasonable number of air changes per hour, taking into consideration the construction and use of the building under consideration (see Table 125).

TABLE 125

APPROXIMATE NUMBER OF AIR CHANGES PER HOUR DUE TO INFILTRATION IN SUMMER

Arrangement of Openings	Estimated Number of Air Changes per Hour
No windows or outside doors	$\frac{3}{4}$
Windows or doors in one wall	1
Windows or doors in two walls	$1\frac{1}{2}$
Windows or doors in three or four walls	2
Stores	2

The preceding table applies to rooms in which the doors are closed practically all the time. In air-conditioned spaces such as stores, banks, and barber shops the estimate of air infiltration should be increased to allow for frequent opening and closing of doors. Infiltration of outside air in cubic feet per minute may be assumed to be 1.3 times the number of persons passing in or out per hour for a 72-in. revolving door and 2 times the number of persons passing per hour for a 36-in. swinging door. If the doors are left open the infiltration may be assumed as 1200 cfm and 800 cfm respectively. If doors must be provided on opposite sides of a building which is to be air conditioned, entryways each provided with two sets of doors should be provided as the infiltration through two doors on opposite sides of a building which are opened simultaneously may exceed 10,000 cfm.

In tall buildings which are air conditioned in summer there may be an appreciable reversed stack effect because the air inside has a lower temperature and consequently a greater weight per unit volume than the outdoor air. The greater density of the column of air in the building causes the inside pressure on the lower floors to be greater than that outside. There is, therefore, a tendency for the conditioned air to flow out through any cracks in the lower part of the structure and for it to be replaced by an inflow of warm outdoor air through cracks in the upper portion.

Infiltration of outdoor air may be neglected when large amounts of ventilating air are used and the conditioned space has relatively few windows, provided that there is no mechanical exhaust system. In this case, a pressure slightly greater than atmospheric is maintained in the rooms that are served by the system.

Sensible heat and moisture gains due to infiltration of outdoor air into the conditioned space must be estimated separately from the same items for outdoor air that is drawn through the air-conditioning machine. The cooling and dehumidifying coil or washer is required to remove the excess sensible heat and the excess moisture from all the fresh air entering the rooms served, but the required condition of the air delivered is affected only by the outdoor air which enters by infiltration or is by-passed around the conditioning machine. It is not affected by the fresh air brought in through the cooling and dehumidifying apparatus because the excess sensible heat and the excess moisture of the outdoor air are removed in the air-conditioning unit and are not absorbed by the air delivered to the conditioned spaces by the conditioner. This point will be further discussed in the next chapter.

TABLE 126

PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR*

Group	Material	Permeability, Grains per Sq Ft (Hr) (Inch Hg)
1†	Plaster base and plaster ¾, in.	14.7
	Fir sheathing, ¾ in.	2.9
	Waterproof paper‡	49.1
	Pine lap siding	4.9
	Paint film	3.4
	Sugar cane fiberboard, ¾ in.	12.5
	Brick masonry, 4 in.	1.1
2§	Foil-surfaced reflective insulation, double-faced	0.08 to 0.13
	Roll roofing, smooth, 40 to 65 lb per roll 108 sq ft	0.13 to 0.17
	Duplex or laminated papers, 30-30-30	1.37 to 2.58
	Duplex or laminated papers, 30-60-30	0.52 to 0.86
	Duplex paper, coated with metallic oxides	0.52 to 1.29
	Insulation backup paper, treated	0.86 to 3.42
	Plaster, wood lath	11.00
	Plaster, 3 coats of lead and oil	3.68 to 3.84
	Plaster, 2 coats of aluminum paint	1.15
	Plaster, fiberboard or gypsum lath	19.73 to 20.57
	Plywood, ½-in., 5-ply Douglas fir	2.67 to 2.74
	Plywood, 2 coats of asphalt paint	0.43
	Plywood, 2 coats of aluminum paint	1.29
	Gypsum lath with metallic aluminum backing	0.09 to 0.39
	Insulating lath and sheathing, board type	25.68 to 34.27
	Insulating sheathing, surface-coated	3.03 to 4.36
	Insulating cork blocks, 1 in	6.19
	Mineral wool, unprotected, 4 in.	29.07
	Sheathing paper, asphalt impregnated, glossy	0.17 to 2.05

[*] From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.

[†] "Calculating Vapor and Heat Transfer through Walls," by L. G. Miller, *Heating and Ventilating*, November 1938.

[‡] Light weight slaters felt used to keep rain from drifting through. Not used as a vapor barrier.

[§] "How to Overcome Condensation in Building Walls and Attics," by L. V. Teesdale, *Heating and Ventilating*, April 1939.

390. Moisture Gain through Walls. Atmospheric air is made of dry air and water vapor, each of which exists at its own partial pressure. Even if the total pressure is the same on the two sides of a wall or partition there will be a difference in the partial pressures of the water vapor if the humidity ratio of the air on one side is greater than the same property on the other side. Therefore there is always a tendency for water vapor to pass through the enclosing envelope of an air-conditioned space when the humidity ratio is maintained at a lower level than that of the surrounding air. Unless the walls or partitions, separating the conditioned space from the outside air or from adjoining spaces, include a vapor barrier, there may be an appreciable moisture gain from this source.

The total amount of water vapor transmitted in this manner to a conditioned space is dependent on the vapor-pressure difference and on the permeability of the wall, which is usually expressed in grains of moisture per square foot per hour per inch of mercury pressure difference. Table 126 gives values for the permeability of various materials used in building construction.

In general, the moisture gain by transmission through the walls, floor, and ceiling of the conditioned space would be a negligible portion of the total moisture gain, but for a warehouse where there are few occupants and where little ventilation air is used it may be the principal factor contributing to the latent heat load on the conditioning apparatus.

391. Heat Gain through Duct Walls. Heat gains through duct walls will occur when either supply or return ducts pass through portions of the building which are not cooled. The greatest gain is likely to occur through the walls of the supply duct because of the greater temperature differential. The heat gain from this source may be estimated by means of the following equation.

$$H = UA(t_o - t_i) \tag{153}$$

where H = heat gain, Btu per hr.

U = overall coefficient of heat transfer outside air to inside air (1.3 for uninsulated ducts, 0.41 for ducts insulated with $\frac{1}{2}$-in. cork or equivalent), Btu per hr per sq ft per deg F. U is affected by the velocity of the air stream. The values given are average ones for air-conditioning systems.

A = outside surface area of ducts passing through unconditioned space, sq ft.

t_o = air temperature in the unconditioned space outside of the duct, deg F.

t_i = air temperature inside duct, deg F.

In general, heat gains through duct walls may be neglected, but where long ducts must pass through unconditioned spaces or where ducts pass through an enclosure such as a boiler room or an unventilated attic they should be estimated and included in the total cooling load used as a basis for the selection of the refrigerating equipment.

392. Heat Gain from Fan Work. The energy delivered by the blades of a fan in an air-conditioning system is eventually converted into sensible heat by friction between the moving air particles and the duct walls and by friction between the particles themselves. This heat becomes part of the sensible heat which must be removed by the cooling unit, and if the fan is located in the discharge side of the cooling coil or washer it will also effect the required condition of the air leaving this section of the conditioning apparatus. If bearing friction is neglected, the heat from which may not reach the air stream, the heat gain from fan work is equal to the shaft horsepower times 2545 expressed in Btu per hour.

393. Duct Leakage. Leakage of treated air from supply ducts passing through spaces which are not conditioned must be compensated by the treatment of additional supply air resulting in an increased load on the apparatus. It is said that as much as 30 per cent of the supply air may be lost in this manner where long ducts have been carelessly fabricated. It is not necessary to provide extra refrigerating and air handling capacity to compensate for duct leakage in ordinary systems if good workmanship in fabricating the ducts can be depended upon. However, in systems which include long runs, an allowance of from 5 to 10 per cent should be included for this item depending upon the length of the runs.

394. Survey. In making an estimate of the cooling load to be imposed on an air-conditioning machine it is necessary to obtain all the pertinent facts about the local summer weather, the building or space to be served, and the indoor conditions that will be satisfactory for its usage. If building plans of adequate size are available the necessary information may be entered on them as the survey is made. Several of the large manufacturers of air-conditioning equipment are in a position to furnish their customers with special printed forms which are a convenience in making the survey and in estimating the cooling load.

395. Example of Cooling-Load Calculations. There is no standard procedure which may be used in estimating the cooling load of a proposed summer air-conditioning plant because each individual installation is likely to present problems which are peculiar to it alone. The purpose of the following illustrative calculations is to indicate a method of approach which in a general way is applicable to problems of this type.

Example. Figure 325 shows a plan of a restaurant which is to be equipped with a system capable of maintaining comfort conditions in the serving area during hot weather. The following information was obtained from a thorough preliminary survey. Design conditions: outside, 95 F dbt, 78 F wbt, 72 F dpt, and 47 per cent relative humidity; inside, 80 F dbt, 66.7 F wbt, 60 F dpt, and 50 per cent relative humidity. Building data: location near 40 deg north latitude; ceiling height, 12 ft; outer walls, 8-in. brick; common wall with adjacent store, 8-in. brick; partition between the serving area and the kitchen, 2 by 4 in. studs with lath and plaster on both sides; windows of single-thickness glass, 8 ft high and equipped with canvas awnings; the door will be considered as part of a continuous window; and horizontal roof of 6-in. concrete, single pitch and slag, and insulated. The building includes a basement. Equipment located in the area to be conditioned: three coffee urns, 3-gal capacity each; one 5000-watt toaster; and total lighting load 6000 watts. Miscellaneous information: The restaurant is open from 6 A.M. until 10:00 P.M. The rush hours are from 6:30 to 7:30 A.M., 12 noon to 1:00 P.M., and from 6:00 to 7:00 P.M. The entire seating capacity (80 persons) may be used during the rush hours. One cashier and eight waitresses work in the serving area during those periods.

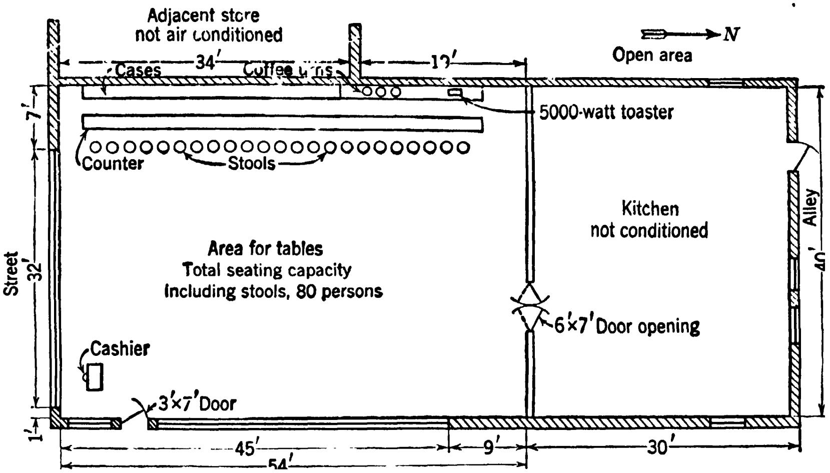


FIG. 325. Plan of restaurant to be air conditioned.

Solution. Because of the large heat and moisture gain from the occupants during the rush hours the peak load on the cooling apparatus is certain to occur either around 12:30 P.M. or around 6:30 P.M. The total heat gain from all sources will be calculated for each of these two periods for August 1. The calculations for the midpoint of the lunchtime rush hour are made as follows:

1. *Sensible-heat gain through east window and door.* (The door will be considered as part of the window.) Since a canvas awning does not appreciably

reduce the heat gain through a glass area in an east wall after 12 noon sun time, Table 116 may be used without correction. Heat gain at 12:30 P.M. = 16.0 Btu per sq ft per hr. $A_1 = 8 \times 45 = 360$ sq ft. $H_1 = 16 \times 360 = 5760$ Btu per hr.

2. *Sensible-heat gain through east wall exclusive of window and door.* Heat gain through 8-in. brick wall with an eastern exposure at 12:30 P.M. from curve 1, Fig. 322, is 3.8 Btu per sq ft per hr. Net wall area $= (12 \times 54) - 360 = 288$ sq ft. $H_2 = 3.8 \times 288 = 1093$ Btu per hr.

3. *Sensible-heat gain through south window shaded with a canvas awning.* Heat gain through single unshaded common glass with a southern exposure at 12:30 P.M. from Table 116 = 89.5 Btu per sq ft per hr. Correction factor due to shading by a canvas awning from Table 119 = 0.3. $A_2 = 8 \times 32 = 256$ sq ft. $H_3 = 89.5 \times 0.3 \times 256 = 6870$ Btu per hr.

4. *Sensible-heat gain through south wall exclusive of window.* Heat gain at 12:30 P.M. from curve 1, Fig. 323, is 1.5 Btu per hr per sq ft. $A_4 = (12 \times 40) - 256 = 224$ sq ft. $H_4 = 1.5 \times 224 = 336$ Btu per hr.

5. *Sensible-heat gain through the portion of the west wall which is exposed to the weather.* Heat gain through 8-in. brick wall at 12:30 P.M. from curve 1, Fig. 324, is 1.5 Btu per sq ft per hr. $A_5 = 12 \times 19 = 228$ sq ft. $H_5 = 1.5 \times 228 = 342$ Btu per hr.

6. *Sensible-heat gain through west wall separating the restaurant from the unconditioned adjacent store.* It will be assumed that the store is well ventilated and that the temperature in the store is the same at all times as that of the outdoor air, Fig. 315. Equation 148 will be used. $H = UA(T_n - T_c)U_6 = \dfrac{1}{\dfrac{1}{1.65} + \dfrac{8}{5} + \dfrac{1}{1.65}}$ $= \dfrac{1}{2.81} = 0.356$. $A_6 = 12 \times 34 = 408$ sq ft. $H_6 = 0.356 \times 408 \times (92 - 80)$ $= 1742$ Btu per hr.

7. *Sensible-heat gain through interior partition separating serving area from the kitchen.* The temperature in the kitchen will be taken as 15 deg above the outdoor temperature at 12:30 P.M. or $92 + 15 = 107$ F. The door will be included as part of the partition. The resistances to heat transfer through this partition are two inside-air films, two separate walls of lath and plaster and an air space between them. $U_7 = \dfrac{1}{\dfrac{2}{1.65} + \dfrac{2}{2.5} + \dfrac{1}{1.3}} = \dfrac{1}{2.782} = 0.359$. $A_7 = 12 \times 40 = 480$ sq ft. $H_7 = 0.359 \times 480 \times (107 - 80) = 4650$ Btu per hr.

8. *Sensible-heat gain through roof.* Heat gain at 12:30 P.M. through one square foot of insulated concrete roof 6 in. thick from curve 8, Fig. 319, is 2.0 Btu per sq ft per hr. $A_8 = 40 \times 54 = 2160$ sq ft. $H_8 = 2 \times 2160 = 4320$ Btu per hr.

9. *Sensible-heat gain from the toaster and the lighting.* The sensible heat liberated by various appliances is given in Table 122. $H_9 = (5000 + 6000) \times 3.416$ $= 37{,}600$ Btu per hr.

10. *Sensible-heat and latent heat gains from three coffee urns* (*each has 3-gallon capacity,* Table 122. $H_{10} = 3 \times 3 \times 1025 = 9225$ Btu per hr of sensible heat. $H_{10} = 3 \times 3 \times 1025 = 9225$ Btu per hr of latent heat.

11. *Sensible-heat and latent-heat gains from inactive occupants* (*customers and cashier*). From Table 124 it is found that the sensible heat emitted by the average persons at rest in an air-conditioned room maintained at 79 F is 225 Btu per hr and that the latent heat emitted is 159 Btu per hr. Although the design inside

temperature is 80 F the data of Table 124 may be used with slight error. H_{11} = 225 × 81 = 18,200 Btu per hr of sensible heat. H_{11} = 159 × 81 = 12,900 Btu per hr of latent heat.

12. *Sensible-heat and latent-heat gain from active occupants (eight waitresses).* It will be assumed that the activity of the waitresses is equivalent to rapid walking under which condition the sensible heat and latent heat emitted per person are found from Table 124 to be 452 Btu per hr and 938 Btu per hr, respectively. $H_{12} = 452 \times 8 = 3616$ Btu per hr of sensible heat. $H_{12} = 938 \times 8 = 7504$ Btu per hr of latent heat.

13. *Sensible-heat and latent-heat gains due to the introduction of outside air for the purpose of ventilation.* Because of the expense involved in conditioning this air, a minimum volume of 10 cfm per person will be used. Assuming standard barometric pressure the volume of one pound of dry air and contained water vapor at 92 F dbt and 77.2 F wbt is 14.27 cu ft per lb. (It is assumed that the specific humidity is the same as under the design conditions of 95 F dbt and 78 F wbt occurring at 3:00 P.M.) The weight of the dry air brought in for ventilation = 10 × 89 × 60 ÷ 14.27 = 3740 lb per hr. Using equations 149 to 152 inclusive. $H_t = W(h_o - h_i) = 3740(40.6 - 31.4) = 34{,}400$ Btu per hr of total heat. $M = \dfrac{W(m_o - m_i)}{7000} = \dfrac{[3740(118 - 76.5)]}{7000} = 22.15$ lb per hr of moisture. $H_L = W \times h_{fg} = 22.15 \times 1050 = 23{,}200$ Btu per hr of latent heat. $H_s = H_t - H_L = 34{,}400 - 23{,}200 = 11{,}200$ Btu per hr of sensible heat.

14. *Sensible-heat and latent-heat gains from the warm food on the tables in the serving area.* From Table 122 it may be found that the sensible heat and the latent heat from this source each amounts to approximately 30 Btu per person. $H_{14} = 30 \times 80 = 2400$ Btu per hr of sensible heat. $H_{14} = 30 \times 80 = 2400$ Btu per hr of latent heat.

Because of the outdoor air introduced for the purpose of ventilation, a slight pressure will be maintained in the conditioned area, and infiltration of outside air may be neglected. Moisture gain through the walls due to the higher vapor pressure on the opposite sides would be a negligible factor in this restaurant because of the large amount of moisture gain from interior sources and in the ventilating air. Since all the area to be served by the machine is in one room there will be no long ducts, and heat gain through the walls of the distributing ducts can be neglected. Likewise, it is not necessary to allow for loss of conditioned air through duct leakage.

By duplicating the foregoing procedure, calculations may be made for 6:30 P.M. The greater of the two totals will be the heat gain used in the design or selection of the conditioning unit. All the pertinent data from the two sets of calculations are given in Table 127.

By a coincidence the estimated total-heat gain for this restaurant is practically the same at 6:30 P.M. as at 12:30 P.M. Some of the heat-gain items are larger at the later hour, but the increase in these items is almost exactly balanced by the decrease in certain others. The maximum heat gain of any room or building which is to be air conditioned can be estimated by application of the same general procedure that was used in the foregoing example. However, the detailed items must be suited to the particular case at hand. Items which are extremely important in one space may be negligible in another used for a different purpose, and vice versa. In making the calculations for certain other types of buildings, it may be necessary to estimate the total heat gain for more than two periods during the day in order to definitely determine the maximum.

TABLE 127

TABULATION OF HEAT GAINS FOR RESTAURANT SHOWN IN FIG. 325

Item Number Referring to Previous Calculations	At 12:30 P.M.		At 6:30 P.M.	
	Sensible Heat, Btu per Hr	Latent Heat, But per Hr	Sensible Heat, Btu per Hr	Latent Heat, Btu per Hr
1	5,760	0	1,080	0
2	1,093	0	2,304	0
3	6,870	0	768	0
4	336	0	1,430	0
5	342	0	1,368	0
6	1,742	0	1,885	0
7	4,650	0	4,825	0
8	4,320	0	10,130	0
9	37,600	0	37,600	0
10	9,225	9,225	9,225	9,225
11	18,200	12,900	18,200	12,900
12	3,616	7,504	3,616	7,504
13	10,200	23,200	10,200	23,200
14	2,400	2,400	2,400	2,400
Totals	106,354	55,229	105,031	55,229
Grand total	161,583		160,260	

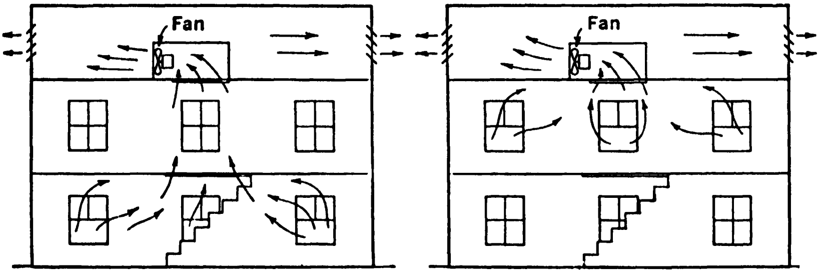
CHAPTER 18

APPARATUS FOR PRODUCING COMFORT IN SUMMER

396. Trends in Summer Air Conditioning. Development of apparatus for producing summer comfort has lagged far behind improvements in heating equipment because summer cooling has been considered less essential than winter heating. Because of the comparatively small number of hours per year that cooling equipment is used and rather high fixed expenses, the cost per hour of comfort produced by such equipment is usually much higher than with a heating system. Usually property damage does not result from the high temperatures of summer, and they do not often constitute a serious threat to human life, so the average home owner with a modest income prefers to suffer some discomfort during the periods of high outdoor temperatures rather than spend the amount of money that would be required for the installation and operation of the equipment that would be necessary for its alleviation.

However, installations for summer air conditioning have paid good dividends in the form of increased sales, more work turned out by employees, or better quality products in many commercial establishments. Equipment for cooling and dehumidifying summer air is now regarded as a necessity by the management of certain types of business enterprises. Home owners who have enjoyed the comfort produced by summer air conditioning while at work or while visiting or shopping are likely to try to find some practical way of bringing the same comfort to their homes. It therefore appears likely that well-designed summer air-conditioning apparatus of all types will find an ever-increasing market for many years.

397. Use of Fans for the Production of Comfort in Summer. Circulation of night air, induced by a suitable fan placed in the attic can bring a considerable measure of relief from the discomfort of hot summer evenings. Figure 326 shows a feasible arrangement and suggests a possible schedule for a two-story home in which the living room is located on the first story and the bedrooms are on the second story. The fan draws comparatively cool night air through the open windows of the rooms that are being used, and the combination of lowered air temperature and increased air movement usually produces a tolerable effective temperature in the path of the flowing air. It is recommended that a fan be selected which has the capacity to provide 20 to 30 air changes per hour or one air change every 2 or 3 min. In a home having an internal volume of 10,000 cu ft, a fan capable of handling between 3000 and 5000 cfm would be required to produce the best results which are possible with this method.

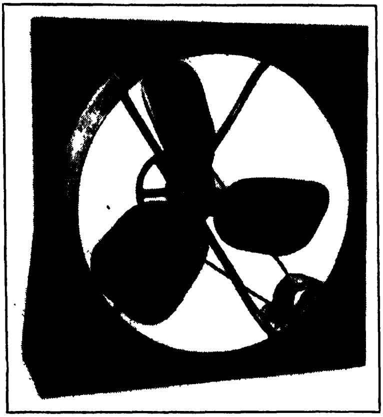


1 st Floor Windows Open During Evening

2 nd Floor Windows Open During Night

Fig. 326. Night-air cooling system.

Several manufacturers are offering fans designed especially for the circulation of night air through homes. Figure 327 shows a fan of this type complete with belt and driving motor. The fan may be placed in a wall of an enclosure over a suitable grille set in the floor of the attic as indicated in Fig. 326, or it may be placed in a gable of the attic. If the gable location is used louvers must be provided at the fan outlet, and all cracks in the roof or between the roof and side walls must be filled to prevent the fan from drawing outdoor air directly into the attic instead of pulling it through the lower portions of the house. Grilles in the floor of the attic should be placed over a hall on the floor below and should be covered with a pad of insulating material during the heating season. A stairway leading to the attic may be used in place of a grille in the attic floor if it happens to be suitably located.



(*Lau Blower Co.*)

Fig. 327. Attic fan for circulation of night air.

If the use of an attic fan is not feasible, circulation of night air through a home or an apartment may be produced by a fan of suitable design placed in a window and directed so as to exhaust air from the room in which it is located. The room selected for the fan location should preferably be the one that is used the least during the evening and night, and all windows in this room must remain closed except the one occupied by the fan. Fan units designed for use in windows can easily be transferred from one window to another. An ordinary 16-in. circulating fan supported so as to face an open window at a distance of approximately 3 ft will exhaust a considerable volume of indoor air and thus induce a flow of night air through the open windows in adjacent rooms.

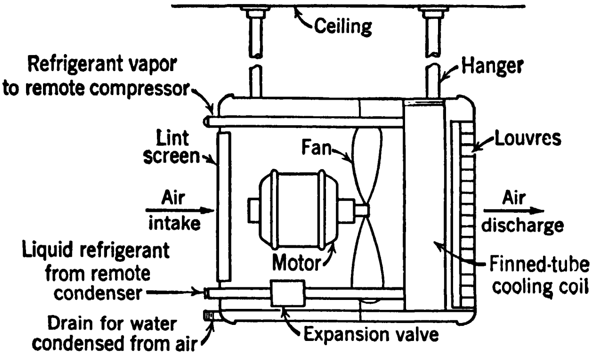
Although circulation of night air is a very practicable method of relieving or at least reducing hot-weather discomfort in homes or other buildings, completely satisfactory summer air conditioning requires the employment of equipment which is much more intricate.

398. Requirements of Summer Air-Conditioning Plants. Summer air-conditioning plants must include one or more fans for producing the required amount of air circulation, a filter for air cleaning, and apparatus for cooling and dehumidification. The majority of summer air-conditioning machines also include a provision for reheating the air when necessary. The incorporation of an odor-absorbing unit, Art. 318, often permits the recirculation of a greater proportion of the air and affects a reduction in operating expense.

The various processes by which air may be cooled, dehumidified, and reheated have been discussed in Chap. 2. Some of the equipment which may be used in summer air-conditioning systems is illustrated and its applications discussed in the following articles.

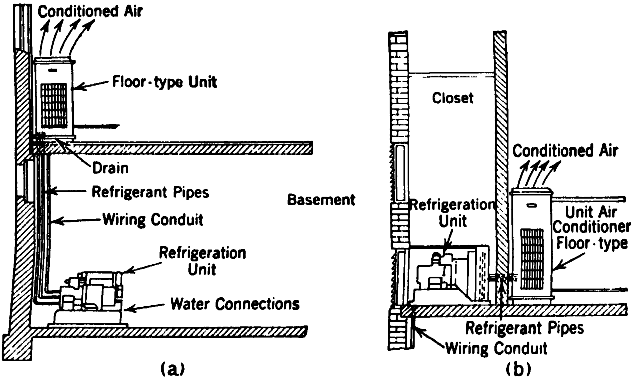
399. Remote Room-Cooling Units. The simplest type of summer air conditioner is the remote room-cooling unit, the principal parts of which are a motor-driven fan and a finned-tube coil. The cooling medium supplied to the coil may be either liquid refrigerant, chilled water or brine from a refrigerated cooler, or simply cold water from either a well or a city water system. Satisfactory results can be achieved with water from either a well or a water main only if it is available at temperatures which do not exceed 60 F at any time. In many communities, limited pumping or pipe-line capacity may make it impractical to use water from the city system for this purpose even where it is available at suitable temperatures. In addition to the principal parts which have been mentioned, remote room coolers usually include some sort of air filter, a set of louvers to direct the flow of the cooled air, and a suitable pan with drain connection for handling the water which is removed from the air stream.

Remote units are made in two distinctly different types. One style is arranged for suspension from the ceiling as shown in Fig. 328 while the other is designed for placement on the floor as shown in Fig. 329. Small units of the ceiling style usually employ propeller-type fans whereas those arranged for the floor position usually incorporate one which operates on the centrifugal principle. The floor-type unit is usually arranged to discharge the cooled air upward and is provided with one or more inlet grillés in the lower part of the cabinet. Units of this style are usually encased in a housing which is given an attractive exterior finish and are commonly used in either homes or offices. Units of the suspended type are well adapted to the cooling of stores, restaurants, and other commercial establishments. Either


Ceiling
Refrigerant vapor to remote compressor
Hanger
Lint screen
Fan
Louvres
Air intake
Air discharge
Motor
Liquid refrigerant from remote condenser
Finned-tube cooling coil
Drain for water condensed from air
Expansion valve

(*From* "*Heating Ventilating Air Conditioning Guide 1948.*" *Used by permission.*)

FIG. 328. Suspended propeller-fan-type unit air conditioner.



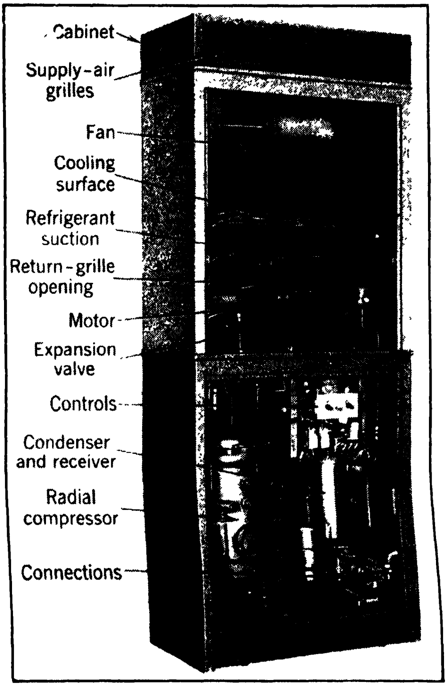
(a) (b)

(*Westinghouse Electric Corp.*)

FIG. 329. Floor-type air conditioner with external refrigerating unit.

style may be obtained in capacities varying from ½ to 2 tons. Both of the units shown in Figs. 328 and 329 are provided with direct expansion coils which must be supplied with liquid refrigerant from a remote condenser. The coils shown must also be connected with the suction side of a remote compressor in order that the refrigerant vapor may be removed. The compressor and the condenser may be located in the basement of the building as indicated in Fig. 329*e* or in an adjacent room or closet as indicated in part *b* of the same figure. If the condenser provided is designed for air cooling, outdoor air is often used for this purpose, and suitable openings in an exterior wall must be provided for the intake and discharge of this air as indicated in Fig. 329*b*. The direct-expansion coil of any remote unit may be replaced by a coil which is designed to circulate either chilled water or chilled brine, provided that the coordinating refrigerating unit is adapted to the cooling of these circulating media. Where cold water is available in adequate amounts and at suitable temperatures all the afore-mentioned refrigerating apparatus may be dispensed with, and only suitable water supply and waste connections are required.

400. Self-Contained Room-Cooling Units. A self-contained room-cooling unit incorporates a small mechanical refrigerating plant and does not require a supply of either liquid refrigerant or chilled water. However, some means must be provided for carrying away the heat that is given up by the refrigerant as it passes through the condenser. This may be accomplished by a stream of outdoor air, handled by a separate fan in an air-cooled condenser, or by means of cooling water in a water-cooled condenser. The air-cooled condenser is the type in most common use in self-contained room-cooling units. Water to be used as condenser



(*Chrysler Corp.*)

FIG. 330. Open view of self-contained room-cooling unit.

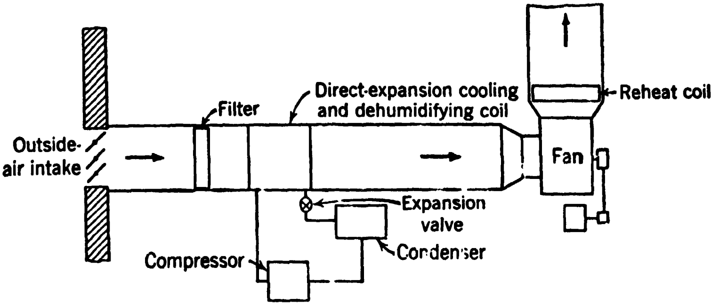
cooling water is not required to be initially as cold as with water for use in the air-cooling coil of a remote unit, and a smaller amount may be used since a higher rise in temperature will not interfere with the results produced.

Figure 330 shows a typical self-contained unit employing a water-cooled condenser and a direct-expansion coil for the cooling and dehumidification of the air.

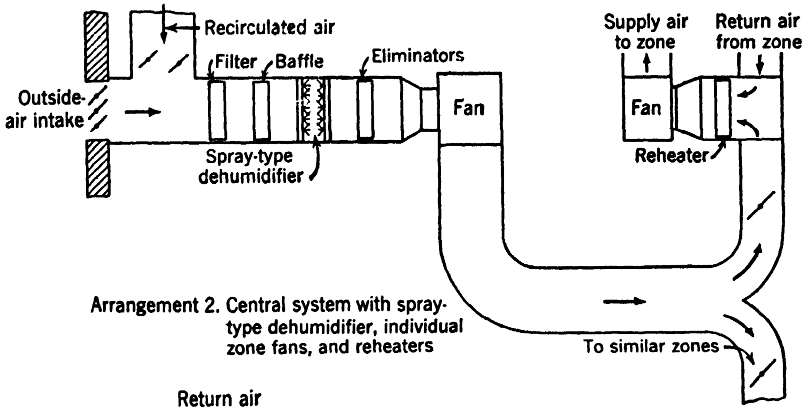
401. Summer Central Systems. A central system for summer air conditioning may employ any one of many different arrangements of apparatus. When the building usage is such as to require 100 per cent fresh air as might be the case with a theater, arrangement 1 in Fig. 331 is the one most commonly used. This arrangement may be modified so as to permit recirculation of part of the air by connecting a recirculating duct into the machine housing between the outside-air intake and the filter. Dampers must be provided in both the outside intake and in the recirculating duct to provide a means of controlling the air proportions. Automatic controls for such a damper combination are discussed in Chap. 20. When the space served by a central system of this type is divided into zones a smaller reheat coil may be installed in each zone-supply duct in place of the one large coil near the fan outlet. The steam or hot-water supply to each coil may then be controlled by a separate thermostat located in the zone served by it. When more than one system is to be installed in a building, installation and maintenance costs may be reduced by substituting a chilled-water coil for the direct-expansion coil in each system, in which case a central water-chilling unit such as the one described in Art. 372, supplies chilled water to all of the air cooling coils in the building. The compressors, condensers, and expansion coils would then be eliminated from the individual units.

Arrangement 2 in Fig. 331 illustrates the use of a chilled-water spray for the cooling and dehumidification of the air supply to all the zones. This arrangement employs a separate fan for each zone and provides for the recirculation of different proportions of air in the different zones. Provision for recirculation within each zone makes use of the by-pass method, which will be discussed in Art. 405.

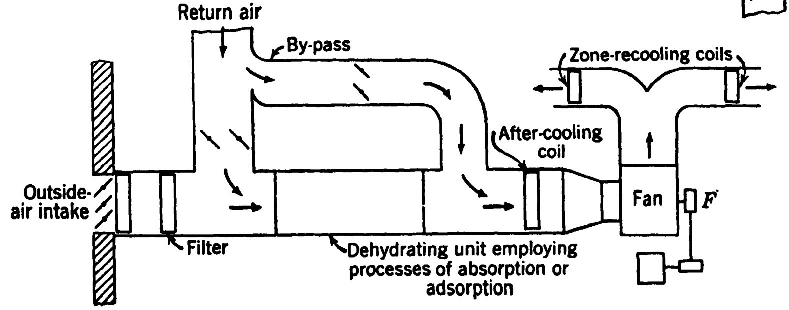
Arrangement 3 in Fig. 331 shows a major digression from the principles of the other arrangements in that dehumidification is by either the process known as adsorption, or that known as absorption, which must be followed by after cooling. The details of the apparatus required to produce dehumidification by absorption or adsorption will be discussed in the following article. The principle advantage of either of these arrangements over the others shown is that it is not



Arrangement 1. Single central system supplying 100 per cent fresh air at same temperature to all the spaces served



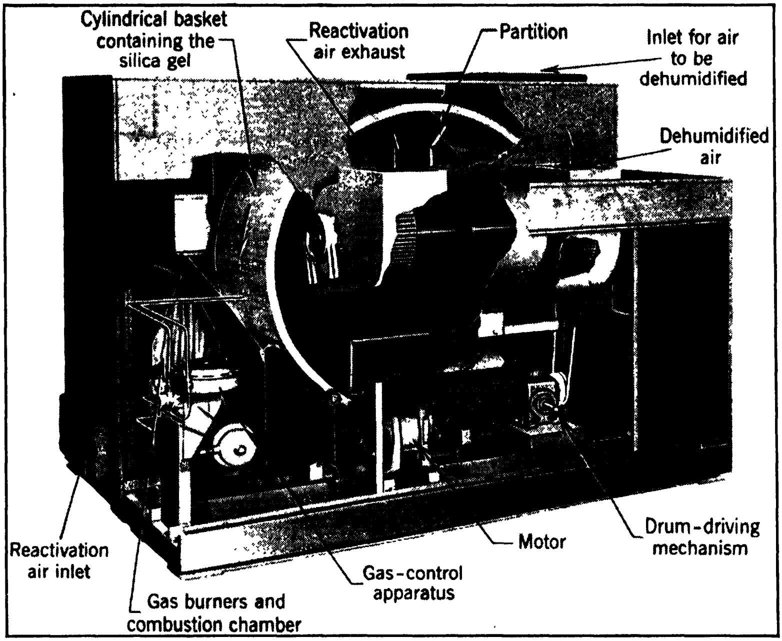
Arrangement 2. Central system with spray-type dehumidifier, individual zone fans, and reheaters



Arrangement 3. Central system in which air is dehumidified by absorption or adsorption and then cooled. Final cooling is controlled by zone thermostats

FIG. 331. Arrangements of apparatus in central summer air-conditioning systems.

necessary to cool the air to a temperature below that desired for entry into the spaces served. Because the air does not have to be cooled to a low temperature to effect the necessary dehumidification, water for the cooling coils may have a higher initial temperature, thus making it practicable to use city water in most cases where it is available. Even where it is necessary to chill the water, a lesser amount is required for a system in which the air is dehumidified in this manner.



(*Bryant Heater Co.*)

FIG. 332. Dehumidifying unit employing silica gel.

402. Apparatus for Dehumidification by Absorption or Adsorption. Special salt solutions capable of absorbing water vapor from air that is brought into intimate contact with them may be used as agents for dehumidification, but since they may also be used for humidification the necessary apparatus for applying this principle will be discussed in Chap. 19, which deals with all-year air-conditioning machines.

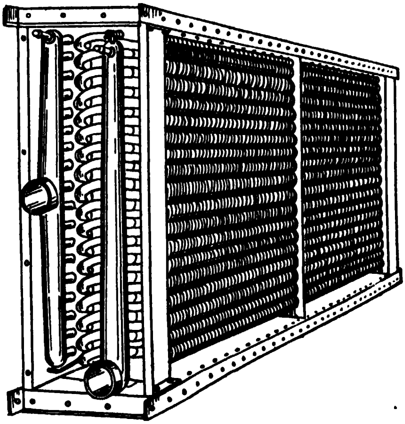
Substances for dehumidifying air by adsorption include silica gel and activated alumina. The process is discussed in Art. 55. Silica gel is a product of fused sodium silicate and sulfuric acid and has the appearance of quartz sand. It is the adsorbent most commonly used in summer air-conditioning machines. Figure 332 shows apparatus which may be used for the dehumidification of air by passing it through a bed of silica gel. Apparatus of this sort must contain a provision for the reactivation of the gel after it has adsorbed moisture equal to approximately 25 per cent of its own weight. In the arrangement shown in Fig. 332 the gel in the form of small particles is contained in a cylindrical basket the interior of which is divided into two compartments. Partitions outside the drum together with the one inside make it possible to direct one stream of air through the gel in one half the basket while a separate stream is directed through that in the other half. The right half of the apparatus shown dries the air that is being conditioned while a stream of air from a heater at a temperature of approximately 350 F is passed through the gel in the left half of the basket to cause its reactivation. In the reactivation process the water which entered the pores of the gel from the air being dried is evaporated into the stream of heated air. The drum is continuously rotated at the rate of approximately one revolution in 7 min. The air stream used for reactivation is usually taken from outside the building and discharged to the outside after it has served its purpose. Many other arrangements for the use of this material in the dehumidification of air are possible, but all must provide for its reactivation by the use of alternate beds, a rotating drum, or the temporary discontinuance of the service of the bed.

Since silica gel reduces the moisture content of a stream of air passing through it to a lower value than is desirable in most rooms, a summer air conditioning machine equipped with a dehumidifying unit employing this principle should provide a means of by-passing a portion of the air around it as indicated in arrangement 3, Fig. 331. The air must be cooled following dehumidification by adsorption before being delivered to the space served because its temperature is increased by this process. Cooling of the air after it has been dehumidified by coming in contact with an adsorber may be accomplished by passing it over a coil supplied with cool water (below 70 F) or over a direct-expansion coil supplied with liquid refrigerant. Another scheme which may be used in certain circumstances for "after cooling" is to pass the warm air with very low moisture content through a spray of recirculated water, in which case the air is cooled by the process of evaporative cooling. The advantage of this arrangement is that the air may be cooled to a temperature considerably below that of the available cooling water without the aid of mechanical refrigeration. When evaporative cooling is to be used for lowering the temperature of the air coming from an adsorber, a greater portion must be passed through the drying unit and a lesser portion by-passed around it.

This method of "after cooling" cannot be applied if a low relative humidity is required in the conditioned space.

The adsorption system of dehumidification has its most advantageous application in certain fields of industrial air conditioning where extremely-low relative humidities are desired in the conditioned space. It may also be applied to advantage in installations where the latent-heat load constitutes an unusually high proportion of the total heat to be removed from the air. Since the moisture is removed in a process that is entirely separate from the removal of sensible heat, almost any desired relative humidity can be maintained in the conditioned space without difficulty or excessive cost, provided that the apparatus is properly designed and equipped with suitable by-pass ducts, dampers, and controls.

In general, absorbent solutions mentioned at the beginning of this article are better adapted to use in air conditioning than are solid adsorbents such as silica gel. However, the solid adsorbents may be used over a greater range of air temperatures and relative humidities because these materials are not subject to a change of state under any of the conditions encountered. They may therefore be applied in certain special dehumidification problems where the absorbent solutions could not be considered.



(Aerofin Corp.)

FIG. 333. Aerofin continuous-tube cooling coil.

403. Types of Cooling Coils. Cooling coils in summer air-conditioning machines may transfer heat directly from air to an evaporating refrigerant, in which case they are called direct-expansion coils, or they may transfer it to chilled water which in turn is cooled by mechanical refrigeration. Direct-expansion coils are discussed in Art. 371. Figure 333 shows an example of a finned-tube coil suitable for inclusion in a summer air-conditioning machine and designed for the use of chilled water as the cooling medium.

Because one heat transfer is eliminated when direct-expansion cooling coils are used it is possible for the refrigerant to evaporate at a higher pressure while effecting a required reduction in the temperature of the air that is being conditioned. Therefore systems which make use of direct-expansion coils cost less to install and operate with less expense since the pressure differential to be maintained by the compressor is lower. However, when several systems are to be installed in one building, there is an important advantage in using chilled-water coils in order that one mechanical refrigerating system may serve the entire building by supplying liquid refrigerant to a central water cooler. Liquid refrigerant from the condenser of a central refrigerating plant may be piped to several different direct-expansion coils in as many different air-conditioning machines, but the danger of leaks developing in long pipes transporting liquid refrigerant causes many designers to specify chilled-water coils instead of direct-expansion coils when more than one air-cooling unit is to be installed in a building. A method of selecting cooling coils is given in Art. 421.

404. Coils for Reheating. Where dehumidification is accomplished by cooling, the air temperature must be reduced to a value that is considerably below that required in the space served. Since the air leaving the washer or coil is usually at least 20 deg cooler than that desired in the conditioned rooms, the air must be reheated after it has been dehumidified unless the sensible-heat gain in the same spaces from all external and internal sources is sufficient to supply the required amount of heat that is needed for this purpose. Machines which are equipped with reheating coils (for example) either arrangement 1 or 2 in Fig. 331 are capable of maintaining accurate control of both temperature and relative humidity in the conditioned space. However, reheating is expensive and may be avoided, at least partially, where accurate control of the relative humidity is not essential. This may be accomplished through control of the volume of conditioned air or through the use of the by-pass method which is discussed in the following article.

Heat-transfer coils suitable for use in a hot-blast heating system may be used for reheating in a central summer air-conditioning machine. A greater range of coil types may be used in reheating than for cooling because no condensation of moisture ever occurs on the outer surfaces of coils used for this purpose.

405. Use of a By-Pass in Central Summer Air-Conditioning Systems. A by-pass as used in a summer air-conditioning machine is an arrangement of ducts equipped with dampers and provided for the purpose of passing a portion of the air around the dehumidifying section. This arrangement, which was formerly covered by patents, is often highly desirable as it affords a means of temperature control without resorting to reheat and while delivering substantially a constant total-air quantity. A by-pass is an essential part of a machine such as the one illustrated in arrangement 3, Fig. 331, in which dehumidification is brought about by means of an adsorbing agent, because without it the relative humidity in the conditioned space would usually be too low, and the cost of cooling the air would be unnecessarily high. For a machine which dehumidifies the air by reducing its temperature below the entering dew point, the final air temperature may be controlled by a thermostatically operated damper in the by-pass, in which case the process is known as automatic by-pass. This method of temperature control must, however, be used with caution as the moisture content of the air supplied to the conditioned space will increase with the proportion that is by-passed. Some reheating may be necessary in order to achieve a satisfactory control of the final relative humidity, but a by-pass may reduce the amount of reheating required without producing a relative humidity that is high enough to cause discomfort to the occupants of the space served. Control of the damper in the by-pass duct by means of an instrument that is sensitive to the relative humidity in the conditioned space is particularly well adapted to a machine utilizing the adsorption process for dehumidification of the air.

406. Zoning in Summer Air Conditioning. When a summer air-conditioning machine serves more than one room, the sensible-and latent-heat gains in the different spaces served are likely to vary in such a manner that different temperatures of the air supplied are required for best results. Zoning consists of an arrangement of equipment such that the temperature of each space or group of spaces served by the system may be individually controlled. Accurate control of the relative humidity in the different zones served by one unit for cooling and dehumidification is impossible, but usually the variation is not enough to affect the comfort of the occupants so that the added expense of providing a separate system for each zone is not justified.

Solar-heat gain is one of the principal reasons why zoning is often desirable because its effect varies with the time of day. Large changes in the occupancy of one room with constant occupancy of another served by the same conditioner is also a situation which calls for zoning. Rooms intended for similar usage at all times which are situated so as to receive similar or negligible sun effect may be served as one zone.

Following is a list including four different methods which may be used for achieving separate zone control with a central unit for cooling and dehumidification.

1. Volume control.
2. Separate reheating when necessary, following dehumidification by cooling.
3. Separate recooling following dehumidification by adsorption or absorption.
4. Multiple fans with individual by-pass.

Volume control is achieved by means of dampers in each of the branch ducts so that the quantity of conditioned air delivered to each room can be varied according to the requirement of the individual spaces at any particular time. The dampers may be air operated by remote control; this arrangement combined with a suitable system of temperature indication makes it possible for the building engineer to maintain satisfactory conditions in all rooms at all times without leaving the control room. This system of zoning is the least expensive and most frequently utilized but its use may result in either insufficient ventilation or unsatisfactory distribution of the air at times.

Zone control by separate reheating is accomplished by omitting the reheat coil from the central apparatus and placing a smaller reheat coil in each of the zone supply ducts. The heat output of each coil is then controlled by thermostat located in the zone which it serves. Suitable temperatures together with satisfactory air distribution can be maintained in all spaces served at all times when this system is properly designed.

Zone control by separate recooling after dehumidification by adsorption or absorption also affords a very satisfactory method of compensating for varying sensible-heat gains in the different parts of the building. When the air is dehumidified in this manner the resulting temperature is always higher than that desired in the air supplied so that a thermostatically-regulated cooling coil in each branch duct can be effective in the control of conditions in the spaces served.

Zone control may be achieved by means of separate fans each one of which receives cool dehumidified air from a central conditioning unit as illustrated in arrangement 2, Fig. 331. Each fan is arranged to recirculate a controlled portion of the air it handles from the zone it serves. In this way the amount of air that is by-passed around the conditioner can be regulated according to the requirements of each individual zone.

Several other methods of zoning are possible, and various combinations of systems may be used. Where wide variations in load occur, volume control can be used to reduce the air quantity a limited amount; then as the load continues to fall off, by-pass, or reheat may be employed to prevent the volume of air delivered from becoming so low as to cause unsatisfactory distribution. Each case should be studied with regard to its particular requirements when deciding on the method of zoning that is to be used.

407. Evaporative Cooling. Evaporative cooling as a method of summer air conditioning is impractical except in climates where the outside wet-bulb depression is high or as a supplement to a drying process (see Art. 402). However, it does provide a very low-cost method of cooling where it is applicable. Air at 100 F dbt bulb and 60 F wbt can be cooled to approximately 62 F dbt by passing it through a spray of recirculated water. Air coming from the washer would be nearly saturated, but under certain circumstances satisfactory conditions may be maintained by introducing this low-temperature moist air in proper amounts into a room in which the sensible-heat gain is high and the latent-heat gain is low.

408. Procedure in Designing a Summer Air-Conditioning Machine. Following is an outline of the customary procedure in designing a central system for summer air conditioning.

1. Select inside design conditions (see Chap. 3).
2. Determine outside design conditions from weather-bureau records (see Chap. 17).
3. Determine ventilation requirements of spaces to be served (see Chap. 14).
4. Estimate the maximum sensible-heat gain and maximum moisture gain in the spaces which are to be supplied with conditioned air (see Chap. 17).
5. Study the layout and uses of the spaces served, and decide upon the most practicable method of zoning, the number of zones required, and the rooms to be served by each zone.
6. Select the equipment and design the duct work (see Chap. 13).

A numerical example will be given in the next chapter for an all-year air-conditioning system which will include all the steps required in the design of apparatus for summer conditioning only.

409. Most Economical Use of an Air Washer for Cooling and Dehumidification, Apparatus Dew Point. When dehumidification of summer air is brought about by passing it through a spray of chilled water the air will leave the washer in practically a saturated state (see Art. 53) and at a dry-bulb temperature that is usually lower than that desired in the conditioned space. If the air leaving the washer is saturated its dry-bulb temperature is also its dew point temperature. In any case the dew-point temperature of the air leaving the dehumidifying chamber, fixes the moisture content which it will have when it enters the room. When the air is delivered from the conditioner its temperature will gradually increase to the equilibrium dry-bulb temperature in the room. As the temperature of the processed air increases it absorbs heat from the air already in the room, thus tending to reduce the room temperature. This tendency of the cool air to reduce the room temperature just balances the tendency for the sensible-heat gains to increase it, and the desired temperature is maintained, provided that the proper amount of air is delivered.

If there were no moisture liberation within the conditioned room and all the air entered through the conditioner, the state point would move horizontally to the right on a psychrometric chart (line *AB* in Fig. 334) as it approaches the constant room temperature represented by point *B*. Under these conditions the correct dew-point temperature for the air leaving the washer would be that corresponding to the specified room condition represented by point *A*. However, if there is moisture liberation within the conditioned space, it will tend to increase the moisture content of the air in the room and raise its dew-point temperature. The only way this tendency can be counteracted is to introduce conditioned air having a lower moisture content and a lower dew-point temperature than that corresponding to the specified room condition so that the absorption of moisture from the air in the room by the air from the conditioner will just balance the moisture gain from occupants and all other sources. When the dew-point temperature of the conditioned air is lower than that of the room air, the state point will no longer move horizontally to the right but will move along a condition line (often called the load-ratio line) which

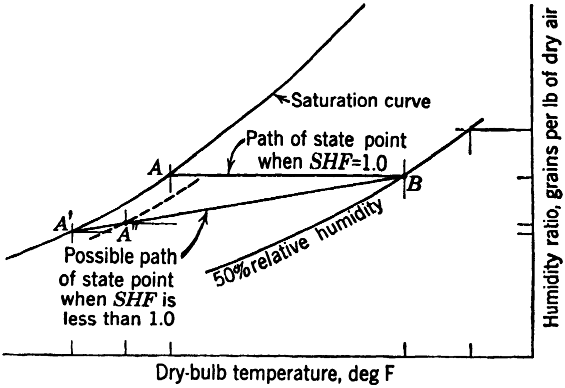

Saturation curve
Path of state point when SHF=1.0
A
B
A'
A''
Possible path of state point when SHF is less than 1.0
50% relative humidity
Humidity ratio, grains per lb of dry air
Dry-bulb temperature, deg F

FIG. 334. Skeleton psychrometric chart showing path of state point or load-ratio line for two conditions of operation.

slopes upward from the horizontal (line $A'B$ in Fig. 334). The slope of this line is determined by the ratio of the sensible-heat gain in the space served to the total of the sensible- and latent-heat gains. This ratio is known as the sensible-heat factor, *SHF*.

$$SHF = \frac{H_s}{H_s + H_L} \tag{154}$$

where H_s = sensible-heat gain of the conditioned space, Btu per hr.
H_L = latent-heat gain of the conditioned space, Btu per hr.

If the condition line crosses the saturation curve, as at A′ in Fig. 334, the point of intersection is called the *apparatus dew point* for the particular problem under consideration.

In the operation of a summer cooling and dehumidifying unit with 100 per cent of the air recirculated the apparatus must absorb an amount of sensible heat that is equal to the sensible heat gain of the space served, and the following relationship must be satisfied.

$$H_s = W_a \times 0.24 \times (t_r - t_e)^1 \tag{155}$$

where H_s = sensible heat gain of the conditioned space, Btu per hr.
W_a = weight of dry air circulated, lb per hr.
t_r = room temperature, deg F.
t_e = temperature of air entering room from the conditioner, deg F.

Assuming an average latent heat of vaporization of 1050 Btu per lb of water vapor condensed, the relationship expressed by equation 156 must also be satisfied.

$$H_L = \frac{W_a(w_r - w_e)}{7000} \times 1050 \tag{156}$$

where H_L = latent heat gain of the conditioned space, Btu per hr.
W_a = weight of dry air circulated, lb per hr.
w_r = humidity ratio of air at specified room conditions, grains per lb of dry air.
w_e = humidity ratio of air entering room from conditioner, grains per lb of dry air.

It is immediately apparent that the problem of controlling a summer air-conditioning machine so as to maintain a specified room condition in the most economical manner is not an easy one to solve.

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[1] The sensible heat absorbed by the contained water vapor is neglected in this equation.

If equilibrium conditions are to be maintained, the ratio of the sensible heat to the total heat absorbed by each pound of air from the conditioner must equal the sensible-heat factor for the entire area served. The sensible heat absorbed by each pound of dry air = $0.24(t_r - t_e)$ (neglecting the sensible heat absorbed by the contained water vapor). The total heat absorbed per pound of dry air = $h_r - h_e$. Therefore

$$SHF = \frac{0.24(t_r - t_e)}{h_r - h_e} \tag{157}$$

where SHF = sensible heat factor for the space served.
t_r = room temperature, deg F.
t_e = temperature of air entering room from conditioner, deg F.
h_r = enthalpy of one pound of dry air and contained moisture under room conditions, Btu per lb of dry air.
h_e = enthalpy of one pound of dry air and contained moisture as it enters the room from the conditioner, Btu per lb of dry air.

If the air leaving the washer is saturated and there is no reheating, t_e is the same as the dew point of the air leaving the conditioner and is also equal to A' Fig. 334, the apparatus dew-point temperature. It is highly desirable to operate in this manner if possible because in this way the desired room temperature is maintained without resorting to reheating, and the cost of operation is held to a minimum.

The correct apparatus dew point for any given installation operating without reheat may be found by estimating the sensible heat factor, *SHF*, then finding by trial and error a value for t_e that will cause equation 157 to be satisfied for the specified condition to be maintained in the room.

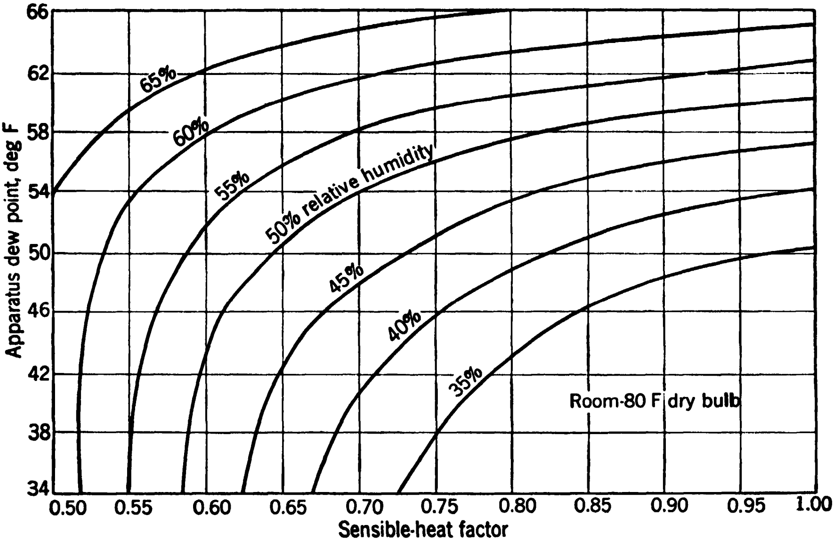
Example. The air in a room is to be maintained at 80 F dbt and 50 per cent relative humidity by 100 per cent recirculation through a washer. The estimated sensible-heat gain under design conditions is 75,000 Btu per hr, and the maximum moisture liberation within the space from all sources is estimated to be 167,000 grains per hr. Find (*a*) the sensible-heat factor, (*b*) the correct apparatus dew-point temperature, and (*c*) the weight of air that must be recirculated per hour, assuming that it is delivered in a saturated state at the apparatus dew-point temperature.

Solution. (*a*) The latent-heat load = (167,000 × 1050) ÷ 7000 = 25,000 Btu per hr, and *SHF* = 75,000 ÷ (75,000 + 25,000) = 0.75.

(*b*) The apparatus dew point selected must satisfy equation 157, $SHF = 0.24(t_r - t_e) \div (h_r - h_e)$. When $SHF = 0.75$, $t_r = 80$ F, and h_r at 80 F and 50 per cent relative humidity = 31.2 Btu per lb, equation 157 becomes 0.75 =

$0.24(80 - t_e) \div (31.2 - h_e)$, using an apparatus dew-point temperature of 60 F for which $t_e = 60$ and $h_e = 26.4$ (saturated air at 60 F), $SHF = 0.24(80 - 60) \div (31.2 - 26.4) = 4.80 \div 4.8 = 1.00$, which is too high. With an apparatus dew-point temperature of 50 F for which $t_e = 50$ and $h_e = 20.3$, the equation becomes $SHF = 0.24(80 - 50) \div (31.2 - 20.3) = 7.2 \div 10.9 = 0.660$, which is too low. A trial with 55 F for which $t_e = 55$ and $h_e = 23.2$ gives $SHF = 0.24(80 - 55) \div (31.2 - 23.2) = 6.0 \div 8.0 = 0.75$, which is the required value of 0.75. Therefore, the necessary apparatus dew-point temperature is 55 F.

(c) The weight of conditioned air required may be calculated most conveniently by dividing the total-heat gain of the space served by the total heat absorbed by one pound of dry air and contained moisture. Therefore the required weight equals $100{,}000 \div (31.2 - 23.2) = 12{,}500$ lb per hr.



(*From "Heating Ventilating Air Conditioning Guide 1947." Used by permission.*)

FIG. 335. Apparatus dew point for various sensible-heat factors.

Unfortunately there is no way in which a solution of equation 157 may be made without resorting to the trial-and-error method as given in the preceding example. Figure 335 has been prepared by solving equation 157 for a range of relative humidities from 35 to 65 per cent and for sensible-heat factors from 0.50 to 1.00. Figure 335 is applicable only for a room-air temperature of 80 F, but similar charts could be prepared for use in cases where other room temperatures are required for some specific purpose other than the provision of comfort for human occupants. From a study of Fig. 335 it may be observed that it is not always possible to select the proper apparatus dew point from this chart when the required relative humidity is low and the sensible-heat factor is also low. For example, the vertical line representing an *SHF* equal to 0.55 does not intersect the curved line representing a constant relative humidity of 50 per cent when the room air temperature is maintained at 80 F. This means that the condition line *A'B* in Fig. 334 is so steep that it does not intersect the saturation curve. In a situation of this kind, which might occur in a restaurant or in any conditioned space where the latent-heat load is unusually high in proportion to the gain in sensible heat, it would be necessary to select a combination of dew-point temperature and weight of circulating air that would absorb the moisture that is liberated within the conditioned space. It would then be necessary to reheat the air to whatever extent is required to maintain the desired dry-bulb temperature. The procedure can best be illustrated by means of a probem.

Example. A certain space receives 50,000 Btu of sensible heat per hr under design conditions and gains 333,000 grains of moisture in the same period. It is desired to maintain a condition of 80 F dbt and 50 per cent relative humidity. Specify the required operating conditions assuming 100 per cent recirculation.

Solution. The latent-heat load = (333,000 × 1050)/7000 = 50,000 Btu per hr, the *SHF* is found to be 50,000/100,000 = 0.50, and from Fig. 335 it is noted that saturated air cannot be used to absorb this combination of sensible- and latent-heat loads while maintaining the specified room condition. Assume 40 F as the lowest practicable temperature for the air leaving the washer; then the weight of air required to be circulated in order that the moisture liberated within the space may be absorbed without exceeding the specified relative humidity is found by using equation 156. $H_L = W_a(w_r - w_e)1050 \div 7000$, $H_L = 50{,}000$, w_r at 80 F and 50 per cent relative humidity equals 76, and w_e at 40 F and 100 per cent relative humidity equals 36; then $W_a = (50{,}000 \times 7000) \div (76 - 36)1050 = 8750$ lb per hr. The required temperature leaving the reheat coil may then be determined by the application of equation 155. $H_s = W_a \times 0.24(t_r - t_e)$ or $50{,}000 = 8750 \times 0.24(80 - t_e)$, and solving for t_e the final temperature after reheating is found to be 56.2 F.

The entire foregoing discussion applies only to a case where 100 per cent of the room air is recirculated. However, the slope of the condition line on the psychrometric chart is not affected by the introduction of outdoor air as long as the condition of the air entering the room from the apparatus is held constant. Likewise, the required apparatus dew point is not affected by varying proportions of outdoor air as long as all the air entering is brought through the washer. Introduction of outdoor air instead of using 100 per cent recirculation would affect the load on the cooling unit. Under outdoor design conditions the load on the cooling unit would be made greater by an increase in the proportion of outdoor air. Regardless of the proportion of outdoor air that is used the total load on the cooling and dehumidifying apparatus is given by the following equation.

$$H_c = W_o h_o + W_r h_r - (W_o + W_r)h_a \qquad (158)$$

where H_c = heat to be removed by cooling apparatus, Btu per hr.
W_o = weight of outdoor air, lb of dry air per hr.
h_o = enthalpy of the outdoor air, Btu per lb of dry air.
W_r = weight of air recirculated, lb of dry air per hr.
h_r = enthalpy of air at specified room conditions, Btu per lb of dry air.
h_a = enthalpy of air leaving cooling apparatus. Btu per lb of dry air.

When there is no reheating of the cooled air h_a is also the enthalpy of the air entering the rooms served.

It has been shown that both the sensible-heat gain and the latent-heat gain can usually be taken care of simultaneously by cooling the air to the proper temperature. However, in the actual operation of a summer air-conditioning plant, the *SHF* is likely to vary from hour to hour, thus requiring frequent manual readjustment of the equipment to maintain the required variation in the required slope of the condition line since no automatic-control system has yet been devised that will handle this problem. The usual practice is to select the proper apparatus dew-point temperature for an average value of the *SHF* or for a value that is likely to exist the greater part of the time that the equipment is in operation. At all times when the *SHF* is that for which the apparatus dew-point temperature was chosen, the specified combination of room temperature and relative humidity can be maintained regardless of the magnitude of the total load. With the temperature of the air leaving the washer fixed, the room temperature can be thermostatically regulated by controlling the volume of conditioned air that is delivered. If it is desired to introduce the same volume of air into the room when the total load is less than under design conditions, a by-pass should be used as explained in Art. 405.

Referring to Fig. 336, load-ratio line XY may be assumed to be the one that is correct for the normal or average *SHF* for which the apparatus dew-point temperature X has been chosen. If the *SHF* is decreased owing to an unusually high proportion of latent-heat load or an abnormally low proportion of sensible-heat load the specified room temperature can still be maintained by controlling the volume of cooled and dehumidified air that is delivered. However, it is no longer possible to maintain the specified relative humidity with the temperature of the air still fixed at X. The air delivered in the state represented by point X is able to absorb the amount of moisture $W_y - W_x$ while changing from the saturated state at X to the specified temperature and relative humidity represented by point Y. However, the new load-ratio line takes the state point of the entering air along the line XY', and the increase in moisture content is $W_y' - W_x$, resulting in a relative humidity that is greater than that specified. If the *SHF* becomes greater than normal the load-ratio line will have a lesser slope than XY, as for example, line XY'' in Fig. 336. Each pound of entering air would then absorb less moisture than under the conditions for which the appratus dewpoint was chosen, and the relative humidity in the room will be less than that which was specified. A decrease in relative humidity below that usually specified would not be objectionable from the standpoint of human comfort. The only objection to a room condition at Y'' instead of at Y is that the apparatus dew point X is lower than that required, resulting in operating costs that are higher than necessary. Increased relative humidity as at Y' may be objectionable from the standpoint of human comfort. This condition can be avoided by reheating the saturated air from the washer to the point X'. This could be accomplished automatically by controlling the flow of heating medium to the reheat coil by a humidity-sensitive instrument in the recirculating duct. If the relative humidity in the recirculation duct, which is the same as that in the room, becomes too high reheating would be started, and the heat absorbed by each pound of entering air would be decreased. This would soon result in a demand for more conditioned air by the room thermostat, and the increase in air supply would then result in absorption of the excess moisture.

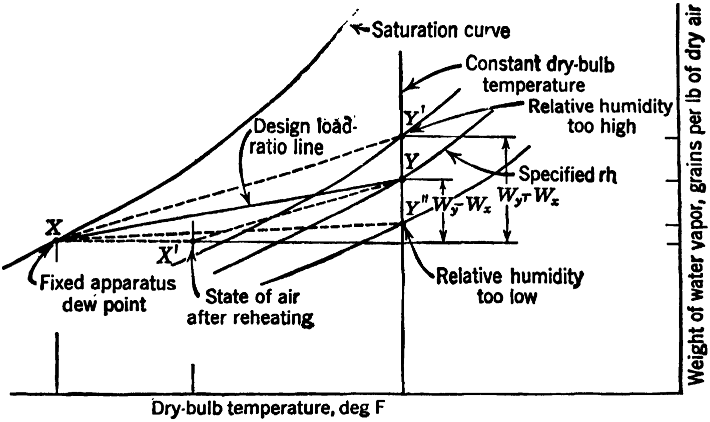


FIG. 336. Skeleton psychrometric chart showing effects of varying load ratio with a fixed apparatus dew point and control of relative humidity through reheating.

410. Most Economical Use of a Coil for Cooling and Dehumidification. The operation of a cooling and dehumidifying system is somewhat different when a coil is used instead of a washer. As was pointed out in Art. 54, the entire air stream does not contact the surfaces of a coil so that a saturated condition is not achieved. Assuming that the broken line in Fig. 334 represents the equilibrium relative humidity reached by the air passing through the coil and that the condition line determined by the SHF is $A'B$, the proper temperature for the air leaving the coil would be A'' instead of A'. Because of the higher air temperature and the higher moisture content at A'' the heat and moisture absorbing capacity of each pound of air circulated would be slightly reduced, and the weight of air circulated would have to be proportionately increased.

The required temperature of the entering air could be determined by a trial-and error solution of equation 157 as in the example that has been worked out for saturated air from a washer, except that h_e for the trial values of t_e will be based on the assumed relative humidity leaving the coils instead of an an assumed saturated condition. The actual relative humidity of the air leaving a coil operated with a surface temperature that is below the dew point of the entering air will depend on the depth and construction of the coil and on the velocity of the air. The assumed relative humidity should be based on actual test data for the coil to be used. After the proper temperature for the air leaving the coil has been determined, the actual coil-surface temperature required to maintain it can be automatically regulated by a control system whose temperature-sensitive element is placed in the air stream at this location.

PROBLEMS

1. A summer air-conditioning machine equipped with a chilled-water spray chamber for cooling and dehumidification is to maintain 70 F dbt and 70 per cent relative humidity in a print shop which has an estimated sensible-heat gain of 50,000 Btu per hr under design conditions. Assume that all of the air is recirculated and that the maximum moisture gain is 100,000 grains per hour. Find (*a*) the latent-heat gain, (*b*) the sensible-heat factor, (*c*) the correct apparatus dew-point temperature, (*d*) the hourly weight of recirculated air, and (*e*) the required refrigeration capacity in tons.

2. Find the weight of air which must be circulated and the temperature to which it must be reheated when the moisture gain of the building of the preceding problem is 400,000 grains per hour and the air is cooled to a minimum temperature of 40 F.

3. Compute the required refrigeration capacity, in tons, when 50 per cent of the air supplied by the machine of problem 1 is fresh air entering it at the outdoor design conditions of 100 F dbt and 82 F wbt.

4. The spray chamber of the machine of problem 1 is to be replaced by a direct-expansion coil which will discharge air having a relative humidity of 90 per cent. Find the temperature at which the air must leave the coil if it is to absorb the sensible- and the latent-heat gains of the space. Find the required weight of processed air necessary when all of the air handled is recirculated.

CHAPTER 19

ALL-YEAR AIR-CONDITIONING METHODS AND EQUIPMENT

411. Requirements of a Machine for All-Year Air Conditioning. All-year air-conditioning machines must be capable of maintaining a specified temperature and relative humidity within the spaces they serve regardless of the outdoor weather conditions. In other words, a machine of this type must perform winter air conditioning which usually involves heating and humidification and also summer air conditioning which generally includes cooling and dehumidification. In addition to the apparatus necessary for properly conditioning the air and distributing it through the spaces served, most all-year air-conditioning machines include filters for dust removal. Many also include special equipment for removal of odors. The term comfort air conditioning applies to spaces in which human comfort and economy of operation are the only considerations. Industrial air conditioning applies to all spaces used for the carrying on of processes which require special air conditions for best results. In industrial air conditioning, human comfort may be of secondary importance.

412. Applications for All-Year Air-Conditioning Machines. All-year air conditioning is desirable for practically every building intended for human occupancy. High initial cost of the necessary equipment has prevented it from becoming economically feasible for the average home owner, but it is regarded as a necessity by the persons in charge of many research laboratories. Control of the temperature and relative humidity is necessary in the manufacture of many products. Table 128 shows optimum temperatures and relative humidities for several industrial processes.

The competition of others in the same business who have already installed all-year air-conditioning machines is forcing many commercial establishments to invest in equipment of this nature. Following are a few examples from a long and ever-growing list of applications: theaters, restaurants, hotels, department stores, offices, classrooms, factories of many types, laboratories, railroad cars, buses, storage chambers, and private homes.

Although the cost of apparatus for all-year air conditioning is greater than that of a heating plant or a summer cooling unit it is usually

TABLE 128

OPTIMUM DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR CERTAIN INDUSTRIAL PROCESSES*

Industry	Process	Optimum Dry-Bulb Temperature, Deg F	Optimum Relative Humidity, Per Cent
Automobile	Assembly line	65 to 80	40 to 55
	Precision parts maching	75 to 80	40 to 55
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Dough retarding	32 to 40	76 to 85
	Paraffin paper wrapping	80	55
	Flour storage	65 to 75	55 to 65
Biological products	Blood bank	38 to 42	60 to 65
Brewing	Fermentation in vat room	44 to 50	50
Ceramic	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
Drug	Deliquescent powder	75	35
	Storage of powders and tablets	70 to 80	30 to 35
	Tablet compressing	70 to 80	40
	Packaging	80	40
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Ripening of meats	40	80
	Storage of citrus fruits	32	80
	Storage of frozen meats	0 to 5	85
	Storage of sugar	80	35
Fur	Storage of furs	28 to 40	50 to 65
Incubators	Chicken hatching	99 to 102	55 to 75
Library	Book storage	65 to 70	38 to 50
Munitions	Fuse loading	70	55
Paint	Air drying lacquers	70 to 90	25 to 50
	Air drying of all paints	60 to 90	25 to 50
Paper	Storage of paper	75 to 80	40 to 60
Photographic	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
Printing	Binding	70	45
	Folding	77	65

TABLE 128 (*Continued*)

Industry	Process	Optimum Dry-Bulb Temperature, Deg F	Optimum Relative Humidity, Per Cent
Rubber	Dipping of surgical rubber articles	80 to 84	25 to 30
	Cementing	80	25 to 30
Soap	Drying	110	70
Textile	Cotton spinning	60 to 80	50 to 70
	Rayon spinning	70	85
	Silk spinning	75 to 80	65 to 70
	Wool spinning	75 to 80	55 to 60
Tobacco	Cigar and cigarette making	70 to 75	55 to 75

* Extracted from Table 1 pp. 836 and 837, "Heating Ventilating Air Conditioning Guide 1947." Used by permission.

considerably less than the combined cost of separate systems for winter heating and summer cooling. It is usually more economical, more convenient and more satisfactory to install an all-year air-conditioning machine, instead of two separate systems. If summer air-conditioning is contemplated for the near future, all-year air conditioning should be carefully considered at the time a building is built or remodeled.

413. Possible Combinations of Processes in All-Year Air Conditioning. An all-year air-conditioning machine may be designed to deliver properly conditioned air during winter weather through employment of the following combinations of processes.

1. (*a*) Preheating to a dry-bulb temperature approximating the dew-point temperature of the desired room condition, (*b*) humidification by means of a spray of heated water maintained at the same temperature, and (*c*) reheating to whatever temperature is necessary in order to maintain the desired room temperature.

2. (*a*) Preheating to a wet-bulb temperature approximating the dew-point temperature of the desired room condition, (*b*) humidification by a spray of water that is neither heated nor cooled, and (*c*) reheating as in combination 1.

3. (*a*) Preheating as in combination 1 followed by humidification through controlled evaporation from an open pan of boiling water or by the introduction of a controlled jet of steam from properly designed nozzles, then reheating as in combinations 1 and 2.

4. (*a*) Preheating when necessary to a temperature well above freezing, and (*b*) humidification through bringing the air in contact with a suitable solution of lithium chloride or calcium chloride maintained at the proper temperature and density, followed by heating or cooling when and as required. Provision must be made for the

regeneration of the absorbent solution by the addition of water to replace that which is given up to the air.

In mild climates where the temperature of the air entering the machine is never lower than 32 F, preheating of the air may be eliminated from combinations 3 and 4.

During summer weather the following combinations of processes may be used.

1. Cooling to the proper temperature followed by reheating if necessary to maintain the desired dry-bulb temperature and relative humidity in the space served. The cooling may be accomplished by passing the air over finned-tube coils or through a spray of chilled water. The air is dehumidified in the cooling process, provided that its temperature is lowered below that of the dew point corresponding to its original state.

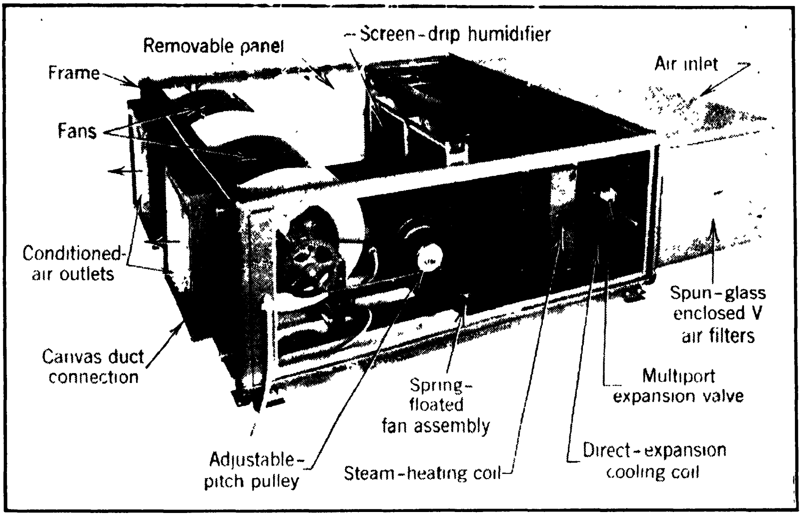
2. Dehumidification by passing the air through a bed of adsorbent material followed by cooling to whatever temperature is required to maintain the desired comfort in the room. The adsorption of moisture from the air raises its temperature, but it is not necessary when using this combination to reduce the air temperature to such a low figure as when combination 1 is used. Reheating of the air is eliminated, but the adsorbent material must be regenerated by the application of heat.

3. Dehumidification by bringing the air in contact with a suitable hygroscopic solution followed by cooling if required. Lithium-chloride solution is the absorbent most commonly used for this purpose. The solution is regenerated by heating to drive off the water vapor absorbed from the air.

The thermodynamics of all of the above summer and winter processes have been discussed in Chap. 2.

In designing an all-year air-conditioning machine it is necessary to select a combination of apparatus that is capable of performing one sequence of processes during winter weather and another sequence of processes in summer. Certain parts of the apparatus that are essential during summer operation will be inoperative when the weather is cold, and vice versa. However, all the air handling equipment and some of the processing equipment can be used at all times. Specific arrangements will be discussed in later articles.

414. Advantages of All-Year Air-Conditioning Machines over Two Separate Systems. The principal advantage in all-year air-conditioning units over two separate systems, one for use in winter and the other for use in summer, is that much of the apparatus, such as inlet grilles, outlet grilles, filters, fans, ducts, and dampers, can be used during both seasons, thereby avoiding costly duplication of equipment. Reheating is required after humidification in winter and under some circumstances after dehumidification in summer so that the same coil or coils and the same control mechanism can be used to perform this function throughout the entire year. It is better for equipment to be in use the year round because apparatus that is idle for several months gathers dust, loses oil from bearings, etc. When humidification is accomplished by bringing the air into intimate contact with water or a suitable solution, much of the humidifying apparatus may be used for air dehumidification in summer. A system capable of processing either summer or winter outdoor air in maintaining comfort conditions in the spaces served will usually occupy considerably less floor area in the building than would be the case if two separate units were installed.



(*General Electric Co.*)

FIG. 337. Unit assembly air conditioner with top and side panels removed.

415. Unit All-Year Air Conditioners. A unit all-year air-conditioning machine is one in which all of the necessary apparatus for heating, humidification, cooling, and dehumidification are included in a single cabinet along with one or more fans and a filter. A remote room-cooling unit, Art. 399, may be converted to a unit all-year air conditioner by the addition of a heating coil and a humidifier. Remote units which are equipped with a water-type coil may be arranged to cool air in summer and heat it in winter with the same coil, supplied with either cold or hot water. Figure 337 shows a remote type of all-year air conditioner which is made in capacities up to 33 tons, and Table 129 includes data in regard to units of the type shown. Units of this type may be obtained with a chilled-water cooling coil in place of the one shown. The steam coil shown for heating may be replaced by one adapted to the circulation of heated water. Self-contained air-conditioning units incorporate a small refrigerating plant and may be classified in several ways as follows: (1) according to the method of rejecting the condenser heat, i.e., air cooled, water cooled or cooled by evaporation; (2) according to the way ventilation air is supplied by the unit, i.e., either one-hundred per cent recirculation with no

TABLE 129

RATING AND OTHER DATA FOR GENERAL ELECTRIC COMPANY REMOTE-TYPE ALL-YEAR AIR CONDITIONERS

Air Conditioner Model Number	Range of Air Flow, Cfm	Range of Cooling Capacity			Range of Heating Capacity		Humidifying, Capacity, Lb per Hr	Face Area, Sq Ft
		Freon 12		Water, Btu per Hr	Steam, Btu per Hr	Hot Water, Btu per Hr		
		Tons	Btu per Hr					
HD-200	1200 to 2400	2.82 to 8.45	34,000 to 102,000	29,000 to 99,000	62,000 to 156,000	75,000 to 182,000	15	4.06
HD-300	2300 to 4700	5.38 to 16.5	65,000 to 199,000	57,000 to 192,000	121,000 to 302,000	145,000 to 353,000	30	7.85
HD-400	3500 to 7100	8.1 to 24.8	98,000 to 300,000	86,000 to 288,000	182,000 to 455,000	218,000 to 530,000	48	11.8
HD-500	4600 to 9400	10.8 to 33.0	130,000 to 398,000	114,000 to 384,000	242,000 to 604,000	290,000 to 706,000	48 to 96	15.7

Rating conditions:

Cooling:
Entering air—84 F dbt, 68 F wbt
Freon 12—40 F, 12 F superheat
Water—44 F

Heating and Humidifying:
Entering Air—70 F dbt, 30 per cent relative humidity
Steam—2 psig
Hot water—180 F entering

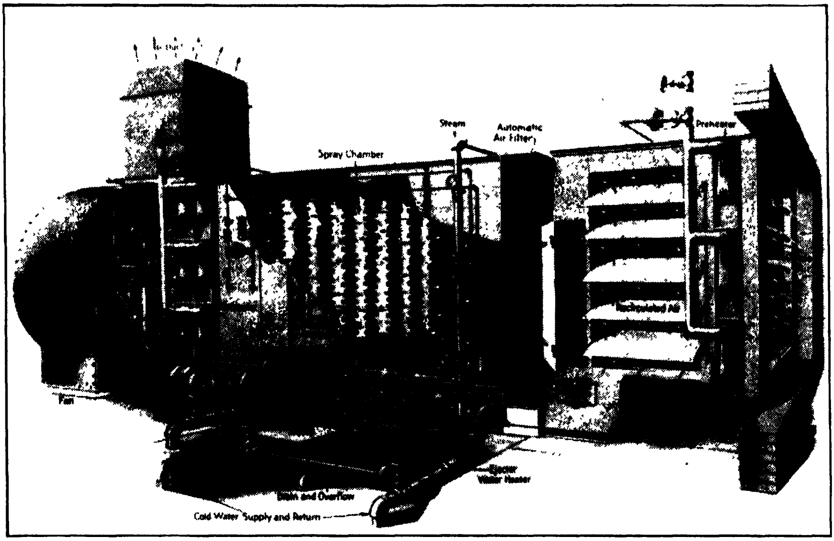
deliberate ventilation, ventilation by drawing air from outside and discharging it into the room, or a combination of outside air and recirculated air; (3) according to the type of air delivery, i.e., free delivery with no ducts from the unit and forced circulation through distributing ducts; or (4) according to the placement of the unit in the room, i.e., floor type or suspended type. Unit all-year air conditioners may be very similar to self-contained room-cooling units such as the one shown in Fig. 330. Some manufacturers offer heating coils as optional equipment which may be added if desired to adapt the machine to use during cold weather. Humidifiers are also offered as an optional extra by some of the makers so that their units may be arranged to perform the functions of a complete winter air conditioner.

The principal advantages of unit conditioners over central systems, which will be discussed later, are reduction in initial cost through standardization and mass production and reduction in installation cost. The engineering involved in making an installation of unit conditioners consists of (1) estimating the cooling and heating loads, (2) choosing the proper types of units, (3) selecting the proper sizes, and (4) arranging them in the proper locations.

Unit systems are especially advantageous in buildings such as hotels where there are a great many different rooms, each one occupied by a person having an individual idea as to what specific conditions provide a comfortable environment. Self-contained units are in general more expensive to operate than either remote units or central systems.

416. Central All-Year Air-Conditioning Plants. When several rooms in the same building are intended for uses which require air having approximately the same temperature and relative humidity, they can usually be air conditioned more economically from a central system than from a number of self-contained units. The operating costs of a central system are usually lower, and the installation cost is usually less, particularly if the system can be installed at the time the building is erected. One of the important advantages in a central system is that the peak summer load does not usually come at the same time in all the spaces served so that the capacity of the cooling and dehumidifying apparatus can be considerably lower than the combined capacity that would be needed if an individual self-contained unit were used in each space.

A great many different arrangements of apparatus may be successfully used to perform the functions of a central all-year air-conditioning machine. Because of space limitations only the ones in general use can be discussed in this chapter. Figure 338 shows one of the simplest arrangements for a central system and provides a positive control of the dry-bulb temperature and relative humidity of the air delivered. The arrangement of inlet dampers permits taking any desired proportion of the air from outside, the remainder being drawn from the conditioned space for recirculation. The apparatus shown in Fig. 338 is designed to employ the processes of preheating, humidification with heated spray water, and reheating during cold weather. Summer air is cooled and dehumidified by chilled spray water in the same chamber that is used for humidification in winter, and the steam flow to the reheater coil may be controlled by a room thermostat to provide just the right amount of reheating during both seasons. The recirculated air enters the machine from the equipment room which serves as a plenum chamber. Suitable arrangements must be made for conducting the air to be recirculated from the spaces served to this room.

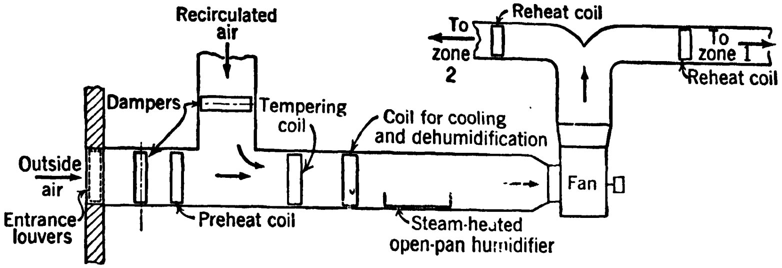


(*Carrier Corp.*)

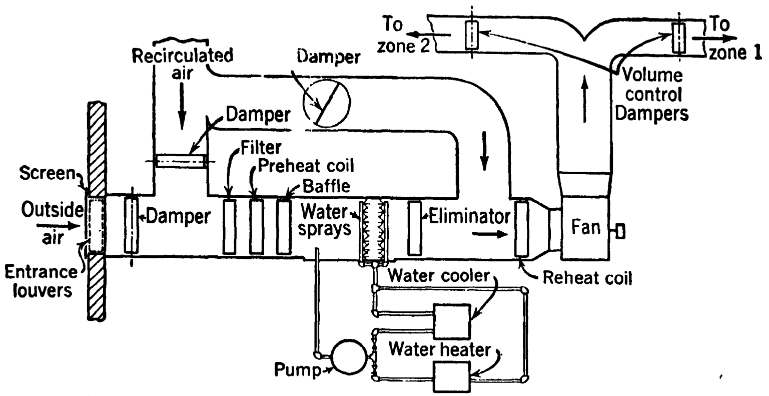
FIG. 338. All-year central air-conditioning system.

Figure 339 shows schematic diagrams of three different arrangements of all-year air-conditioning apparatus which may be used when the building served must be divided into zones in order that satisfactory conditions may be maintained at all times in all of the conditioned areas.

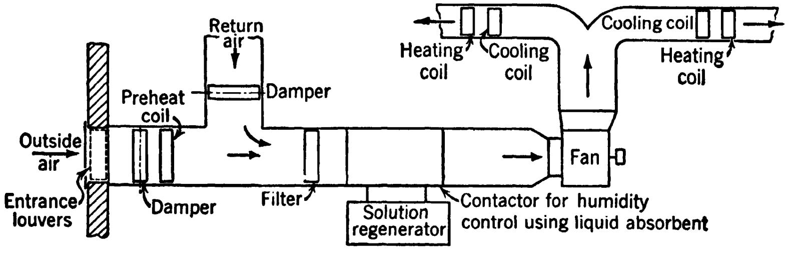
Arrangement 1 provides a positive control of the dry-bulb temperature in the different zones by using a separate reheat coil in each branch of the system instead of one in the central machine as is used in the



Arrangement 1. Central system using separate units for humidification and dehumidification, and an individual reheat coil for each zone



Arrangement 2. Central system using water spray for humidity control. Air delivered to all zones at essentially the same temperature



Arrangement 3. Central system using absorbent solution for humidity control. Zoning by separate coils for final heating or cooling

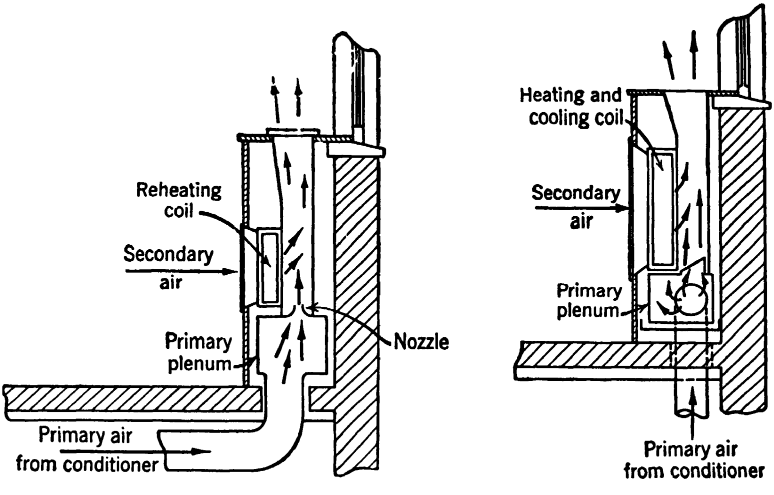
FIG. 339. Various arrangements of apparatus in all-year central systems.

unit of Fig. 338. The combination of a preheat coil and a tempering coil shown in Arrangement 1 of Fig. 339 is necessary in some installations where ventilation requirements may demand that a large proportion of outdoor air be used even in extreme cold weather. The first steam-heating coil in a stream of air at a very low temperature must be "fully on" or "completely off" to prevent freezing of the condensate in the bottom. In order to achieve a satisfactory control of the air temperature entering the humidifying section of the conditioner under those conditions it is then necessary to provide a preheat coil of low capacity which may be operated with its steam valve wide open followed by a tempering coil, the flow of steam to which may be thermostatically controlled. The open-pan type of humidifier indicated in this arrangement and shown in Fig. 11 is inexpensive in first cost and will provide satisfactory humidification if properly controlled. The water level in the pan is maintained by a float-controlled valve, and evaporation into the air stream is accomplished by heating the water by means of steam coils laid in the bottom of the pan. The flow of steam to the coils in the humidifying pan is usually controlled by a humidistat located in the return-air duct. Dehumidification during summer operation is accomplished by means of a finned-tube coil which may be supplied with either chilled water or a liquid refrigerant.

Arrangement 2, Fig. 339, utilizes some form of air washer for both humidification and dehumidification as in Fig. 338. When humidification of the air is required the spray water is heated, and when the outdoor conditions call for dehumidification the spray water is cooled. Either process may be followed by reheating, and volume dampers are used to regulate the delivery of conditioned air to the various zones served by the system. Heat losses in the different zones during winter operation are usually taken care of by radiators when the zoning is by volume control. A discussion of the various types of modern air washers is given in Chap. 14, and the by-pass method which is used in this arrangement is discussed in Art. 405.

Arrangement 3, Fig. 339, includes the same apparatus for proportioning outdoor and recirculated air as was used in Arrangements 1 and 2, but in this case the air is humidified or dehumidified as the occasion demands by bringing it in contact with an absorbent solution which is at all times maintained at the proper temperature and per cent concentration. Provision must be made as indicated for continuous regeneration of the solution. Further discussion of the possibilities of this method of humidity control will be given in a later article.

417. Use of Dampers in Air-Conditioning Machines. Since some air-conditioning machines may serve several rooms whose individual requirements may fluctuate independently, a damper may be required in each branch of the distribution system in order that the volume of air delivered to each room may be varied. When several rooms are air conditioned by one machine, a considerable reduction in operating costs may be effected by the proper use of dampers. This is particularly true if some of the rooms are used a greater proportion of the time than others served by the same system. Dampers for varying the proportion of fresh air are usually necessary if comfort conditions are to be maintained with minimum operating expense. A mixing damper suitable for controlling the proportions of fresh air and recirculated air is shown as part of the apparatus of Fig. 338. Mixing dampers of this type may also be used for controlling the proportion of recirculated air which is by-passed around the cooling and dehumidifying section of an air-conditioning machine.



(*a*) Low-pressure type (*b*) High-pressure type

(*From "Heating Ventilating Air Conditioning Guide 1948." Used by permission.*)

FIG. 340. Induction-type room air conditioners.

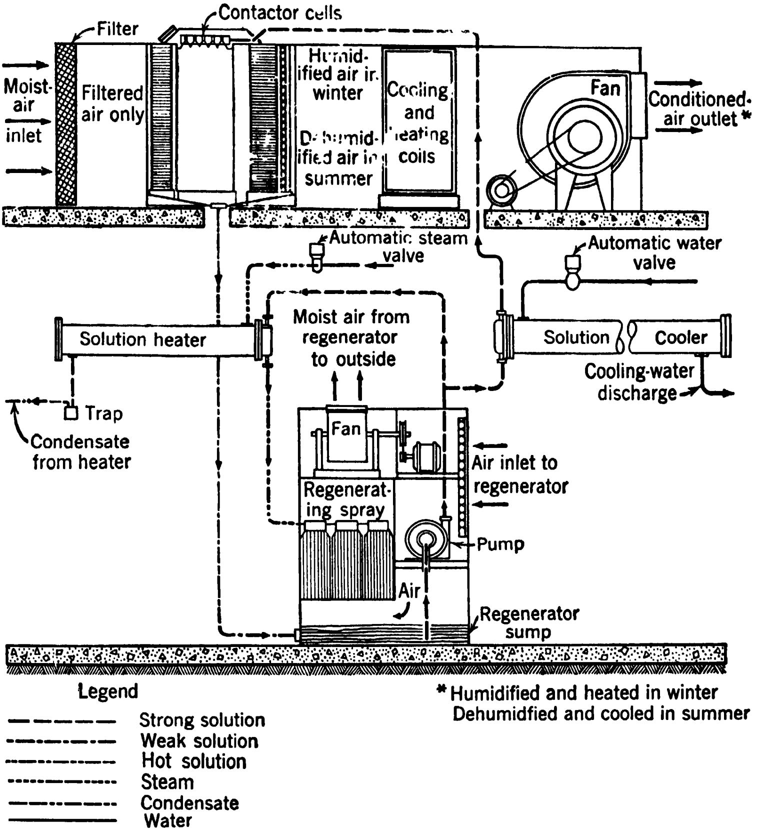
418. Induction Units. Induction units such as either of the two illustrated in Fig. 340 combine the principles of the central system and that of self-contained units. A unit consisting of a cabinet enclosing a primary-air nozzle, a secondary-air inlet from the room, a coil for heating or cooling the secondary air, a mixing tube, and an outlet diffuser is placed in each room. The plenum chamber of each unit is supplied with primary air which has been properly humidified or dehumidified from a central air-conditioning machine. If the heat losses from rooms are small it may be possible to maintain reasonable temperatures by gravity circulation of the secondary air over the heating coil when the central supply system is not in operation. If necessary conventional radiators under separate thermostatic control may be used to care for the heating load when the air conditioner is inoperative.

An advantage in the induction system is that a given volume of air from the conditioning apparatus will cause a much better circulation of air in the rooms served than would be the case with other types of air delivery because of the high rate of induction of secondary air into the stream leaving the air nozzle. Another advantage is that the admixture of secondary air with the conditioned air in summer operation raises the temperature of the delivered air to near room level before it is discharged from the outlets, thus reducing the chance that cold drafts will be noticed by the occupants of the room. Also a minimum amount of cold air is transported through the ducts of the central system, thereby reducing power costs below the amount that would be required if the full delivery of the diffusers were handled by the central fan.

Control of induction units during summer operation is usually through either manual or automatic control of the primary air. It may be advisable to by-pass some air around the cooling and dehumidifying apparatus of the central system during periods of low cooling load so that the volume of primary air in the induction units will not be reduced below the point where satisfactory circulation is produced in the individual rooms.

Induction units usually produce more noise than well-designed diffusers commonly used in other types of air-conditioning systems so that they should not be specified for spaces where the usage requires that the noise level be kept to a minimum. A recent development known as a high-pressure type of induction unit is claimed to have greatly reduced the noise level below that produced by other types. The reduction in noise level is accomplished through the employment of special nozzles for creating a high-velocity jet, noiselessly.

419. The Use of Absorbents in Central All-Year Air-Conditioning Systems. Solutions of certain salts such as lithium chloride or calcium chloride are very strongly hygroscopic and can therefore be used to dehumidify an air stream when it is brought into intimate contact with them. An absorbent salt solution of appreciable concentration will have a water-vapor pressure that is lower than the vapor pressure of pure water at the same temperature. Consequently a spray of such a solution or a surface wetted with it will dehumidify air having a given water-vapor pressure at a higher temperature than would be the case if dehumidification were accomplished by means of a spray of chilled water. As long as the vapor pressure of the solution is less than that of the air, water vapor from the air will enter the solution, causing the air to be dehumidified and the water content of the solution to be increased. Addition of water to a given amount of a solution will lower the concentration of salt and increase the vapor pressure so that regeneration of the solution by boiling off the accumulated water is necessary if the same degree of dehumidification is



(*Surface Combustion Corp.*)

FIG. 341. Air-conditioning apparatus employing a salt solution for humidity control.

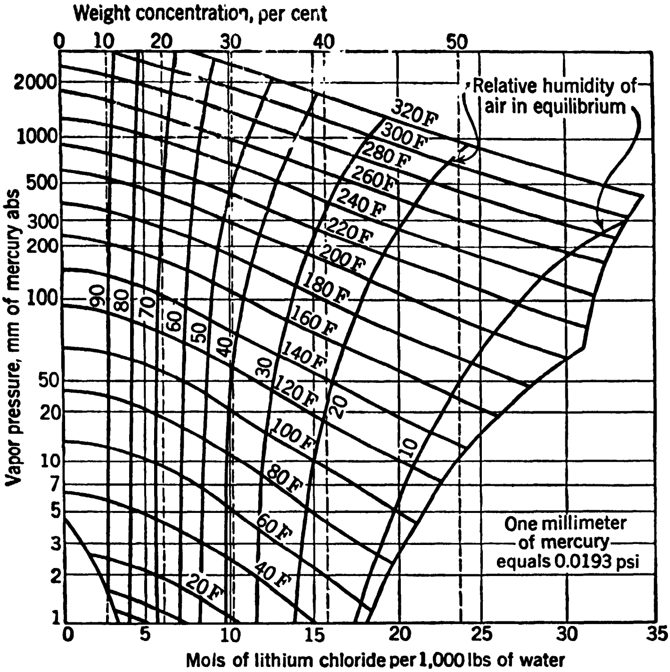
to be maintained. Likewise the latent heat that is released from the water vapor condensed raises the temperature of the solution so that in addition to regeneration by removal of the water accumulated the solution used in the contact chamber must be circulated through a cooler to maintain the proper temperature.

Figure 341 shows the operation of an air-conditioning unit which makes use of a salt solution in the control of the moisture content of the air delivered. The flow diagram is for the processing of summer air. In the arrangement illustrated the pump draws brine from the sump at the bottom of the regenerating chamber and forces approximately 85 per cent of it through a cooler on the way to the spray nozzles above the contactor cells. The automatic water valve regulates the flow of cooling water through the cooler so as to maintain the temperature of the solution entering the contact chamber at the level needed to produce the desired results in regard to the moisture content of the air that is being conditioned. The concentration of the solution is maintained by passing the remaining 15 per cent through a heater, then through the regenerating spray where the heated solution gives up water vapor to an auxiliary air stream which carries it out of the building. By heating the brine that is delivered to the contact chamber instead of cooling it, the air may be humidified by the same unit. When used for humidification it is not necessary to use the regenerating portion of the apparatus, but water must be continually added to the solution to compensate for that which is given up to the air in the humidification process. At all times the brine flows from the contactor cells to the regenerator sump by gravity action.

Lithium chloride has many properties which make it a satisfactory material for use in humidity-control apparatus of the type that has been described. It is less corrosive than clear water, it is perfectly stable, it does not react with the CO_2 in the air, it does not evaporate at the temperature required to regenerate the solution, it does not crystallize in any concentrations that are needed, and it is non-toxic. Figure 342 gives the vapor pressure for different concentrations of this salt in water when the solution is held at different temperatures ranging from 20 F to 320 F. The chart also gives the relative humidity of air in equilibrium with this solution for any combination of temperature and concentration covered. By reference to this chart it is possible to specify a combination of solution concentration and solution temperature for any air condition which may be desired. For example, air which has come to equilibrium with a 40 per cent solution held at 80 F will have a relative humidity of approximately 17 per cent. Likewise, air in equilibrium with a 30 per cent solution held at 100 F will have a relative humidity of 40 per cent.

Absorbent solutions such as lithium chloride are preferable to adsorbent solids such as silica gel when used for humidity control in all-year air-conditioning machines because they may be used for humidification in winter as well as for dehumidification in summer. Even if the adaptability of the solutions to humidification is disregarded they offer the advantage of controlled dehumidification without resorting to by-passing. When used for dehumidification, absorbent solutions afford certain process advantages over cooling coils or sprays of chilled water which have been discussed in Art. 51.

Packaged units employing a solution of lithium chloride for humidity control are available in several different sizes. These units can be adjusted to deliver air having any desired moisture content between wide limits. When packaged units of this type are used they are usually adjusted to deliver air at or near the temperature that is desired in the spaces served. The sensible-heat load in winter is then handled by separately controlled radiators or convectors or by a central hot-blast heating system, and the sensible-cooling load in summer is handled by a separate cooling unit or by a cooling coil in the same system. With a suitable coil and the proper arrangement for either cooling or heating the circulating water, the same heat exchanger may be used to add sensible heat or to remove it as the occasion demands. One advantage of this type of system is that there



(*From "Refrigerating Data Book, 1942." Used by permission.*)

FIG. 342. Physical properties of lithium chloride.

will be no condensation of water vapor on the surface of the cooling coil, thus eliminating the necessity for drainage provisions and also eliminating any chance of objectionable odors from bacterial development on a wetted coil surface.

Lithium-chloride solutions are claimed to be effective in removing certain types of odors which may be present in air as it reaches the contact chamber. It is also claimed that certain air-borne bacteria are rendered harmless after passing through a spray chamber or air scrubber that is supplied with this solution.

The cooling capacity required in an after cooler following an absorbent type of dehumidifier is much lower than when a chilled coil surface or a chilled-water spray is used for cooling and dehumidification of the air. In comparing the cost of operation of the two basically different systems for summer air conditioning, the cost of cooling and regenerating the absorbent solution must be added to the cost of after cooling and the sum compared with the combined cost of dehumidification and reheating in the system which is considered as being conventional at the time of this writing. Proponents of the absorption system claim that a detailed analysis will frequently indicate a substantially lower operating cost when that system is used.

420. Design Calculations for an All-Year Air-Conditioning Machine. A procedure, which in general is applicable to the calculations that must precede the selection of equipment, will be demonstrated by an example. The data in the following example pertain to the same restaurant for which an estimate of maximum heat gain was made in Art. 395. Calculations will first be made for summer operation of the unit because the operating conditions which will produce the desired results with a minimum of expense are less flexible during this season.

Step 1. Selection of temperature and relative humidity which are to be maintained in the conditioned space. The inside dry-bulb temperature and relative humidity most commonly specified in comfort air conditioning are 80 F and 50 per cent respectively.

Step 2. Selection of outdoor dry-bulb and wet-bulb temperatures which are to be assumed. In the restaurant which is to be conditioned these temperatures will be taken as 92 F and 77.2 F respectively.

Step 3. Estimation of maximum heat gains. The maximum heat gain for this restaurant has been estimated in the example of Art. 395 as 161,583 Btu per hr of which 106,354 Btu is sensible heat and 55,229 Btu is latent heat. However, since the ventilation air does not enter directly into the conditioned space, heat gain from this source should not be included in the calculations which are made to determine the required state of the conditioned air. The sensible heat which must be absorbed from the conditioned space by the air entering from the machine = 106,354 − 10,200 = 96,154 Btu per hr, and the latent heat = 55,229 − 23,200 = 32,029 Btu per hr. The total heat to be absorbed by the air from the conditioner = 96,154 + 32,029 = 128,183 Btu per hr.

Step 4. Select the type of unit which is to be used. The sequence of processes to be used in the machine must be decided upon before further calculations can be made. It will be temporarily assumed that the type of unit which will be used is that shown in Fig. 338. In the arrangement shown, the air is cooled and dehumidified by passing it through a spray of chilled water.

Step 5. Calculation of the sensible-heat factor and the apparatus dew-point temperature for summer design conditions. The sensible-heat factor is the sensible heat load divided by the total heat load or 96,154 ÷ 128,183 = 0.75. From Fig. 335 it is found that an apparatus dew-point temperature of 55.0 F is required. The apparatus dew-point temperature may also be found by a trial and error solution of equation 157. (See first example of Art. 409.)

Step 6. Calculation of required weight of dry air to be cooled and dehumidified under summer design conditions, Dividing the total heat gain directly into the space by the heat that will be absorbed by each pound of dry air entering from the machine in a saturated state, at a temperature of 55 F, the weight of dry air to be processed is found as follows: 128,183 ÷ (31.2 − 23.2) = 16,040 lb per hr.

Step 7. Calculation of weights and volumes of air to be handled by the different parts of the system. In the example of Art. 395 it was found that 3740 lb of dry air must be supplied per hr as fresh air to satisfy the ventilation requirements. The specific volume of the fresh air is 14.27 cu ft per lb of dry air, and the volume used equals (3740 × 14.27) ÷ 60 = 890 cfm. The weight of recirculated air to be processed = 16,040 − 3740 = 12,300 lb per hr. The specific volume of the recirculated air is 13.84 cu ft per lb of dry air (assuming standard barometric pressure), and the volume of the recirculated air to be cooled and dehumidified = (12,300 × 13.84) ÷ 60 = 2840 cfm. The state point of the mixture of fresh air and recirculated air may be located on the psychrometric chart by calculating and plotting the enthalpy and specific humidity. After locating the state point of the mixture, the specific volume and dry-bulb temperature may be found as 13.95 cu ft per lb of dry air and 83.2 F respectively. The volume of the air entering the dehumidifying section = (16,040 × 13.95) ÷ 60 = 3735 cfm. The specific volume of the cooled and dehumidified air at a temperature of 55 F is 13.16 cu ft per lb of dry air, and the volume flowing at this point = (16,040 × 13.16) ÷ 60 = 3520 cfm. The cooled and dehumidified air at 55 F is too cool for introduction directly into the conditioned space unless special types of diffusers designed to produce an unusual amount of secondary circulation are used. Provision should preferably be made to mix approximately an equal weight of by-passed air with that which has been cooled and dehumidified.

The volume of the by-passed air would be (16,040 × 13.84) ÷ 60 = 3700 cfm. The dry-bulb temperature and specific volume of the mixture of by-passed air and cooled and dehumidified air may be found to be 67.8 and 13.51 cu ft per lb of dry air.

The fan and the discharge-duct system will then be designed to handle approximately (32,080 × 13.51) ÷ 60 = 7240 cfm. The air which is by-passed does not affect any of the calculations except those in regard to the volume of the air that is handled by the fan and ducts beyond the point where the by-passed air is mixed with the processed air.

Step 8. Calculation of refrigeration capacity required. Equation 158, Chap 18, may be used to find the heat which must be removed from the mixture of outdoor air and recirculated air by the chilled spray water in the washer section. $H_c = [W_o h_o + W_r h_r] - (W_o + W_r)h_a$. $H_c = [(3740 \times 40.6) + (12{,}300 \times 31.2)] - (16{,}040 \times 23.2) = (152{,}000 + 384{,}000 - 372{,}000) = 164{,}000$ Btu per hr. The total cooling load calculated in the preceding manner checks, within a reasonable degree of accuracy, the estimated maximum heat gain for this restaurant at 12:30 P.M. as calculated in Art. 395. A thorough discussion of the operation of a spray chamber for the cooling and dehumidification of air is given in Art. 409. The refrigeration capacity that would be required = 164,000 ÷ 12,000 = 13.7 tons.

Step 9. Selection of indoor temperature and relative humidity for winter operation. Because the windows contain only a single thickness of glass it will not be practicable to maintain the relative humidity above 30 per cent during cold weather. With the relative humidity at 30 per cent a dry-bulb temperature of 72 F is required to produce an effective temperature of 66 F (see comfort chart Art 59).

Step 10. Selection of winter outdoor design temperature. It will be assumed that the lowest temperature on record in the locality of the restaurant is −10 F. The outdoor design temperature will then be taken as 0 F (see Chap. 4).

Step 11. Estimation of heat losses. Following the procedure that has been outlined in Chap. 4 the heat losses due to conduction through exposed wall, window, and roof areas may be estimated at 101,000

Btu per hr. The heat required to raise the temperature of the cold air which enters the restaurant by opening and closing of the door can be neglected because heat gains from people, lights, etc., were not considered in estimating the heat losses. Heat gain from these and other interior sources will be more than adequate for the warming of this air when the serving area is occupied. The heat required to warm the air introduced through the machine for the purpose of ventilation will be provided for in a later step.

Step 12. Calculation of the weights and volumes of air handled under winter design conditions. The weight of the fresh air under winter design conditions (specific volume of outside air, assuming that the air is saturated at 0 F = 11.59 cu ft per lb) when providing 10 cfm per person = (10 × 89 × 60) ÷ 11.59 = 4600 lb per hr. The weight of the total air processed = (3620 × 60) ÷ 14.2 = 15,190 lb per hr, assuming that the fan will handle the same total volume as when the system is on summer operation and that the same proportion will be by-passed. (14.2 is a rough approximation of the specific volume of the air after reheating, assuming that it will be reheated to 100 F.) The weight of the recirculated air may then be found as 15,190 − 4600 = 10,590 lb per hr, and the volume of this air = (10,590 × 13.5) ÷ 60 = 2380 cfm.

Step 13. Estimation of the heating capacity required in the preheater coil. The dew-point temperature of the specified indoor air is 38.5 F, and the preheat coil should preferably temper the incoming fresh air so that the temperature of the mixture of fresh air and recirculated air is approximately at this value. The enthalpy of the mixture of 4600 lb of outdoor air and 10,590 lb of recirculated air under design conditions without preheating = [(4600 × 0.8317) + (10,590 × 22.8)] ÷ 15,190 = 16.15 Btu per lb of dry air. This is greater than the enthalpy of saturated air at 38.5 F. Therefore no preheating is required, and the preheater coil may be omitted.

Step 14. Estimation of the amount of steam required for heating the spray water. From the preceding calculation it was found that the enthalpy of the air entering the washer is above that of saturated air at 38.5 F. Therefore no heating of the spray water will be required. When the temperature of the outdoor air is higher than zero more humidification than is specified will occur, and the relative humidity in the restaurant will be higher than 30 per cent. However, under those circumstances, a somewhat higher relative humidity would not be objectionable as the inside-surface temperature of the window glasses would also be higher, and condensation would not occur on them. It would not be necessary to use the water spray during the rush hours because of the water vapor liberated by food, coffee urns, etc. The spray chamber would likewise not be required during mild weather because of the higher moisture content of the ventilating air.

Step 15. Estimation of the heating capacity required in the reheater coil. Assuming that there will be times when there are no interior sources of heat, the reheat coil must have the capacity to heat the air leaving the washer an amount such that it will compensate for the heat losses from the room while maintaining the temperature at 72 F. The total weight of air processed must liberate 101,000 Btu per hr in cooling from the delivered temperature to that of the room. The maximum heat required from each pound of air delivered by the machine = 101,000 ÷ 15,190 = 6.66 Btu. The required enthalpy of the air leaving the reheater coil is then 22.8 + 6.66 = 29.46 Btu per lb. (22.8 is the enthalpy of air at 72 F and 30 per cent relative humidity). The maximum heat output required from the reheat coil is then = 15,190(29.46 − 16.15) = 202,000 Btu per hr. (16.15 is the minimum enthalpy of the mixture of recirculated air and outdoor air from step 13.) The actual temperature of the reheated air before mixing with the by-passed air would be 98.8 F.

Having completed the foregoing calculations the designer is in a position to specify the apparatus which will be needed. Each of the two multiblade dampers used for proportioning the fresh air and the recirculated air should have sufficient capacity to pass the total amount of air supplied to the conditioner as under some weather conditions it may be advantageous to use 100 per cent fresh air, and under certain conditions when the serving area is not in use it may be desirable to recirculate all the air which is processed. Automatic control of damper combinations is discussed in the next chapter. It will not be necessary to provide a damper in the duct which conducts the by-passed air from the room to the point of mixing with the processed air as it is desirable to have the fan deliver approximately the same volume of air at all times, regardless of the cooling or heating load imposed on the unit. The friction loss caused by the various equipment items may be obtained from data published by the respective manufacturers. The design of the duct system, the selection of the fan, and the specification of the fan speed and motor capacity may be computed as in the example of Art. 303.

Figure 343 is a flow diagram which incorporates weights, volumes, temperatures, and humidity ratios in the various parts of the apparatus when operating under summer design conditions.

Other types of air-conditioning machines would be applicable to this restaurant and should be considered. Since a preheater coil is not required because of the large proportion of air recirculated, the installation cost could be reduced by using a remote conditioner similar to the one shown in Fig. 337. Such a unit could be installed on a properly supported false ceiling near the rear of the serving area and arranged to discharge the conditioned air toward the front. A suitable duct for conducting the mixture of recirculated air and fresh air from the point of mixing to the air intake of the unit could be located above a drop ceiling at the rear of the serving area. The logical location for the fresh-air intake of such a unit could be in the west wall adjacent to the partition which separates the serving area from the kitchen. The recirculated air could be removed from the conditioned space through a grille set with its lower edge at floor level in the northwest corner of the serving area. This grille and the vertical duct necessary for conducting the recirculated air to the mixing chamber could easily be masked, for decorative effect. A unit of this type can be arranged for by-passing air around the cooling and dehumidifying section by installing coils which are not as high as the frame of the machine. Air may also be by-passed by replacing a removable panel with a suitable inlet grille. A direct expansion coil is selected for a unit of this type in the next article. The refrigeration unit consisting of the compressor and the condenser would have to be separately located and could be placed in the basement. If it is necessary to recover the condenser cooling water an evaporative condenser (Fig. 305) could be used. If an evaporative condenser were used, its discharge air duct should be located as far as practicable from the fresh-air intake for the air-conditioning unit.

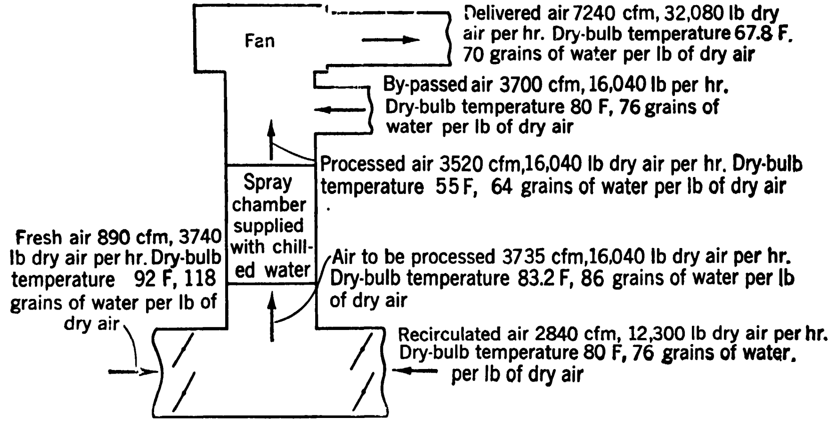

Delivered air 7240 cfm, 32,080 lb dry air per hr. Dry-bulb temperature 67.8 F. 70 grains of water per lb of dry air
Fan
By-passed air 3700 cfm, 16,040 lb per hr. Dry-bulb temperature 80 F, 76 grains of water per lb of dry air
Processed air 3520 cfm, 16,040 lb dry air per hr. Dry-bulb temperature 55 F, 64 grains of water per lb of dry air
Spray chamber supplied with chilled water
Fresh air 890 cfm, 3740 lb dry air per hr. Dry-bulb temperature 92 F, 118 grains of water per lb of dry air
Air to be processed 3735 cfm, 16,040 lb dry air per hr. Dry-bulb temperature 83.2 F, 86 grains of water per lb of dry air
Recirculated air 2840 cfm, 12,300 lb dry air per hr. Dry-bulb temperature 80 F, 76 grains of water per lb of dry air

FIG. 343. Flow diagram for spray-chamber dehumidifier.

The following calculations are made to investigate the possibilities of the absorption type of system shown in Fig. 341. It will be assumed that cooling water is available from a water cooler at 55 F and that the temperature of the lithium-chloride brine can be maintained at 60 F in the contactor cells. It will also be assumed that a 30 per cent concentration will be maintained. From Fig. 342 it may then be found that air in equilibrium with a 30 per cent concentration of this brine at 60 F has a vapor pressure of approximately 5 mm of mercury. This vapor pressure is (5 × 0.0193) = 0.0965 psi. From the psychrometric chart, Fig. 5, it may be found that the humidity ratio of air having a vapor pressure of 0.0965 psia is 29 grains per lb of dry air. Since the humidity ratio corresponding to the specified inside conditions is 76 grains, each pound of dehumidified air is capable of absorbing 76 − 29 = 47 grains per lb of dry air. The maximum latent-heat gain of the conditioned space exclusive of the ventilation air = 32,029 Btu per hr, and the maximum moisture gain = 32,029 ÷ 1050 = 30.50 lb per hr, or 30.50 × 7000 = 213,500 grains per hr. The weight of dry air required from the dehumidifier = 213,500 ÷ 47 = 4540 lb per hr. The weight of dry air to be recirculated with this system under the assumed conditions is then 4540 − 3740 = 800 lb per hr. The heat which must be removed from the mixture of fresh air and recirculated air by the solution in the contactor cells (equation 158) = (3740 × 40.6) + (800 × 31.2) − (4540 × 19) = 152,000 + 25,000 − 86,300 = 80,700 Btu per hr. The dehumidified air must then be cooled to a temperature such that the sensible-heat gain, directly into the space, will be absorbed. The sensible heat which must be absorbed per pound of dry air delivered = (96,154 ÷ 4540) = 21.2 Btu per lb, and the required temperature of the air delivered = 80 − (21.2 ÷ 0.24) = 80 − 88.3 = − 8.3 F. It would not be practicable to cool the air to such a low temperature because frost would form on the cooling coil, and the refrigeration plant could not be operated at its best economy.

Assuming that lithium-chloride brine having the same concentration will be maintained at 80 F in the contactor cells, it is found from Fig. 342 that the vapor pressure of the air leaving would be approximately 11 mm of mercury or 0.212 psia. The humidity ratio of the processed air would then be 64 grains, and each pound of dehumidified air could absorb 76 − 64 = 12 grains, and the weight of dry air to be processed = 213,500 ÷ 12 = 17,800 lb per hr. The required sensible-heat absorption by each pound of processed air = (96,154 ÷ 17,800) = 5.41 Btu per lb, and the required final temperature of the air = 80 − (5.41 ÷ 0.24) = 80 − 22.6 = 57.4 F. The weight of recirculated air would be 17,800 − 3740 = 14,060 lb of dry air per hr, and the heat removed from the mixture of fresh air and recirculated air in the contactor cells = (3740 × 40.6) + (14,060 × 31.2) − (17,800 × 29.3) = 152,000 + 438,000 − 522,000 = 68,000 Btu per hr. This amount of heat would have to be removed from the brine cooler plus the heat of mixing amounting to approximately 100 Btu per lb of water vapor condensed from the air. The heat of mixing would be approximately (213,500 × 100) ÷ 7000 = 305 Btu per hr. The sensible-heat absorption required from the cooling coil = 17,800 (29.3 − 23.8) = 97,900 Btu per hr. The total refrigeration capacity required for this system = (68,000 + 305 + 97 900) ÷ 12,000 = 13.9 tons. In addition to the refrigeration required there would be some expense for the heat required for the brine regenerator. This system would, therefore, be slightly more expensive to operate than either of the other two types on the basis of the assumed brine concentration and conditions of operation. However, there are many possible variations in the use of an absorbent solution for dehumidification of air, and in a large installation the possibilities of the system of Fig. 341 should be investigated by a specialist in its application before deciding on the type of apparatus which is to be used.

421. Selection of Cooling Coils. Before a coil can be properly selected for use as the cooling and dehumidifying section of an air-conditioning unit it is necessary to obtain certain data from actual tests of the available types and to make certain calculations involving the conditions of the specific problem. Several methods of rating coils and of calculating their performances have been proposed. The one presented here is known as the humidity method.[1] In the application of this method to the solution of cooling-coil problems, a psychrometric chart is used together with the following equations.

$$\frac{h_a A_s N}{0.23W} = \log_e \frac{(t_1 - t_{d1})}{(t_2 - t_{d2})} = \log_e R \tag{159}$$

where h_a = air-side film coefficient, Btu per hr per sq ft per deg F. If this is not available from test data it may be approximated from $h_a = 1.1(d_a V)^{0.6}$, where d_a = density of air, lb per cu ft, and V = face area velocity, fpm.

A_s = air-side surface area per square foot of coil-face area per tube row of coil depth, sq ft.

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[1] "Performance of Surface-Coil Dehumidifiers for Comfort Air Conditioning," by G. L. Tuve and L. G. Seigel. *ASHVE Trans.*, p. 523, Vol. 44, 1938.

Air-cooling Coil Problems and Their Solutions, by L. G. Seigel, Heating, Piping and Air Conditioning, *ASHVE Journal Section*, p. 90, February 1945.

N = number of rows of tubes in the path of the air stream.
W = weight of air-vapor mixture per square foot of coil face area, lb per hr.
t_1 = dry-bulb temperature of air entering coil, deg F.
t_{d1} = dew-point temperature of air entering coil, deg F.
t_2 = dry-bulb temperature of air leaving coil, deg F.
t_{d2} = dew-point temperature of air leaving coil, deg F.

$$R = \frac{(t_1 - t_{d1})}{(t_2 - t_{d2})}$$

and

$$t_s = \frac{Rt_2 - t_1}{R - 1} \qquad (160)$$

where t_s = coil-surface temperature, deg F. and the other symbols are as in equation 159.

$$H_t = \frac{h_r}{R_s} A_s N(t_s - t_r) \qquad (161)$$

where H_t = total coil load per square foot of face area, Btu per hr.
h_r = refrigerant-side film coefficient, Btu per hr per sq ft per deg F.[1]
R_s = ratio of air-side surface to refrigerant-side surface.
t_r = refrigerant temperature, deg F. The other symbols are as in equation 159.

The above equations have been developed during a ten-year program of research and may be used for the prediction of air-side performance of cooling coils with an accuracy well within the limits of that possible with psychrometric determinations. The required refrigerant temperature may be estimated within about 2 deg F. The air-side film coefficient h_a can be determined by the application of equation 159

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[1] A value of 325 may be used when the refrigerant is Freon. For either water or brine, under special conditions, consult Fig. 344. For any liquid refrigerant which does not evaporate

$$h_r = 0.0225 v^{0.8} D^{-0.2} d^{0.8} k^{0.6} c_p^{0.4} \mu^{-0.4}$$

where v = velocity, fps.
D = diameter of pipe, ft.
d = density lb per cu ft.
k = conductivity, Btu per hr per sq ft per deg F per ft.
c_p = specific heat, Btu per lb per deg F.
μ = viscosity, lb per hr per ft.

to the data of coil tests. Experience has shown that if the data are taken when the exterior surface of the coil is wet or at least partially wet, the value of h_a obtained may also be applied to the operation of the coil when its exterior surface is dry. This coefficient should be determined for at least three air velocities after which the values may be plotted on log-log paper. The proper value of h_a to be used for predicting the performance of the coil when air at any intermediate velocity is passing over it may be read from the plot. (Air velocities greater than 450 fpm will sweep drops of water from the exterior surface of a coil and should be used with caution.)

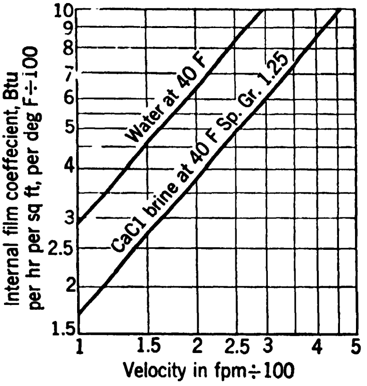


FIG. 344. Refrigerant-side film coefficient h_r when water or CaCl brine is used at a temperature of 40 F in copper tubes with an outside diameter of approximately ¾ in.

Since the air side of the tubes in practically all commercial cooling coils is now provided with fins the area A_s in equation 159 includes the surface of the fins as well as that of the tubes to which they are attached. Because of the finned construction of the coils, the air-side surface temperature is not uniform under any condition of operation. For the purposes of calculation the surface temperature t_s is that of an imaginary thin-walled plain tube for which the air-side heat-transfer coefficient equals that of the actual coil.

When the method is applied to a problem involving dry cooling only (where no dehumidification occurs) the ratio $(t_1 - t_{d1}) \div (t_2 - t_{d2})$ in equation 159 has no significance, but R may still be found from the equation and used in equation 160 to find the required coil surface temperature. The required physical data can usually be obtained from manufacturers' catalogs. Data for one manufacturer's line of direct-expansion coils are given in Table 130.

Example. The method may be applied in the selection of a direct expansion coil to be supplied with Freon, F 12, and used in place of the chilled spray water for cooling and dehumidifying the air in the example of the preceding article. The coil will be designed for a face-area velocity of 400 fpm, and it will be assumed that the proportions of outdoor air and recirculated air will be the same as was used in the previous example.

Solution. The first step will be to locate the state point of the outdoor air and that of the room air on a psychrometric chart, Fig. 345. The state point of the

TABLE 130*

Physical Data for Finned-Tube Direct-Expansion Cooling Coils†

Nominal Tube Length	Net Face Area, Sq Ft			Overall Dimension of Casing, Including 1$\frac{1}{2}$-In. Bolting Flange					
	20$\frac{9}{16}$-In. Unit	29-In. Unit	37$\frac{7}{16}$-In. Unit	For 20$\frac{9}{16}$-In. Units		For 29-In. Units		For 37$\frac{7}{16}$-In. Units	
				Across Tubes	Along Tubes	Across Tubes	Along Tubes	Across Tubes	Along Tubes
2′ 0″	...	3.96	...	...	...	29″	2′ 8$\frac{1}{2}$″	...	...
2′ 6″	3.35	5.01	6.66	20$\frac{9}{16}$″	3′ 2$\frac{1}{2}$″	29″	3′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	3′ 2$\frac{1}{2}$″
3′ 0″	4.06	6.06	8.06	20$\frac{9}{16}$″	3′ 8$\frac{1}{2}$″	29″	3′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	3′ 8$\frac{1}{2}$″
3′ 6″	4.76	7.12	9.46	20$\frac{9}{16}$″	4′ 2$\frac{1}{2}$″	29″	4′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	4′ 2$\frac{1}{2}$″
4′ 0″	5.47	8.18	10.87	20$\frac{9}{16}$″	4′ 8$\frac{1}{2}$″	29″	4′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	4′ 8$\frac{1}{2}$″
4′ 6″	6.18	9.24	12.29	20$\frac{9}{16}$″	5′ 2$\frac{1}{2}$″	29″	5′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	5′ 2$\frac{1}{2}$″
5′ 0″	6.89	10.3	13.68	20$\frac{9}{16}$″	5′ 8$\frac{1}{2}$″	29″	5′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	5′ 8$\frac{1}{2}$″
5′ 6″	7.60	11.34	15.1	20$\frac{9}{16}$″	6′ 2$\frac{1}{2}$″	29″	6′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	6′ 2$\frac{1}{2}$″
6′ 0″	8.30	12.4	16.49	20$\frac{9}{16}$″	6′ 8$\frac{1}{2}$″	29″	6′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	6′ 8$\frac{1}{2}$″
6′ 6″	9.00	13.46	17.9	20$\frac{9}{16}$″	7′ 2$\frac{1}{2}$″	29″	7′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	7′ 2$\frac{1}{2}$″
7′ 0″	9.65	14.48	19.30	20$\frac{9}{16}$″	7′ 8$\frac{1}{2}$″	29″	7′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	7′ 8$\frac{1}{2}$″
7′ 6″	10.35	15.54	20.70	20$\frac{9}{16}$″	8′ 2$\frac{1}{2}$″	29″	8′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	8′ 2$\frac{1}{2}$″
8′ 0″	11.05	16.60	22.10	20$\frac{9}{16}$″	8′ 8$\frac{1}{2}$″	29″	8′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	8′ 8$\frac{1}{2}$″
8′ 6″	11.75	17.65	23.50	20$\frac{9}{16}$″	9′ 2$\frac{1}{2}$″	29″	9′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	9′ 2$\frac{1}{2}$″
9′ 0″	12.45	18.70	24.90	20$\frac{9}{16}$″	9′ 8$\frac{1}{2}$″	29″	9′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	9′ 8$\frac{1}{2}$″
9′ 6″	13.16	19.74	26.30	20$\frac{9}{16}$″	10′ 2$\frac{1}{2}$″	29″	10′ 2$\frac{1}{2}$″	37$\frac{7}{16}$″	10′ 2$\frac{1}{2}$″
10′ 0″	13.85	20.80	27.74	20$\frac{9}{16}$″	10′ 8$\frac{1}{2}$″	29″	10′ 8$\frac{1}{2}$″	37$\frac{7}{16}$″	10′ 8$\frac{1}{2}$″

* Aerofin Corporation.

† General data applicable to all the arrangements listed; Tubing, copper, $\frac{5}{8}$-in. outside diameter. Fins, $\frac{13}{32}$-in. deep, 8 per in. Air-side surface area, 15.95 sq. ft per sq ft of face area per row of tubes. Refrigerant-side surface area, 1.172 sq ft per sq ft of face area per row of tubes. Ratio of air-side to refrigerant-side surface 13.6.

Identical coils, except for refrigerant inlet and outlet piping details are available for chilled water.

mixture entering the coil may then be located after the enthalpy and humidity ratio have been calculated. The sensible-heat factor for the room to be conditioned has been found in Art. 420 to equal 0.75. Equation 157 may be used to locate a point on the load-ratio line. A temperature less than 80 F, for example 60 F, will be assumed and the heat content h_e found that must accompany it if the state point is to be on the required load-ratio line. $0.75 = 0.24(80 - 60)/(31.2 - h_e)$, or $h_e = 24.8$. Therefore the state point of air at 60 F and having an enthalpy of 24.8 Btu per lb of dry air is on the load-ratio line passing through the state of the room air and may be used to establish the slope of this line. It may be noted that this load-ratio line does not pass through the state of the air entering the coil, which means that the coil load-ratio line cannot be the same as the room load-ratio line in this case. A coil should be selected which will remove sensible heat and total heat from the air passing through it in a proportion that will produce a coil load-ratio line which intersects the room load-ratio line at a point near the saturation curve. It will be assumed that the air leaving the coil will have a relative humidity of 90 per cent, and an assumed coil load-ratio line is drawn through the state of the air entering the coil and through the point where the room load ratio line intersects the 90 per cent relative humidity curve on the chart.

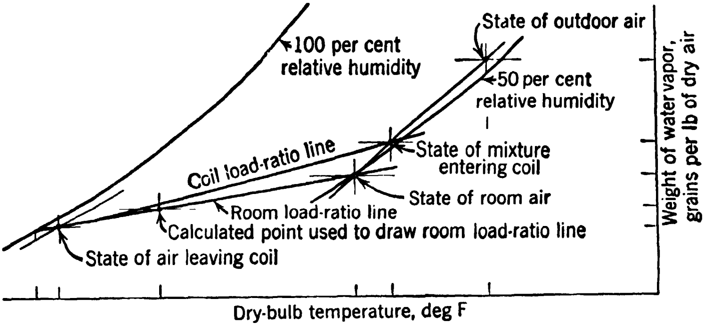

100 per cent relative humidity
State of outdoor air
50 per cent relative humidity
Coil load-ratio line
State of mixture entering coil
State of room air
Room load-ratio line
Calculated point used to draw room load-ratio line
State of air leaving coil
Dry-bulb temperature, deg F
Weight of water vapor, grains per lb of dry air

FIG. 345. Use of psychrometric chart in cooling-coil selection when reheating is not required.

Equation 159 may now be applied to determine the approximate number of rows of tubes required to produce the desired state of the air leaving the coil. $\frac{h_a A_s N}{0.23W} = \log_e \frac{(t_1 - t_{d1})}{(t_2 - t_{d2})}$, h_a, which is assumed to have been obtained from a test of a representative coil of the type selected will be taken as 10.7. A_s, which must be calculated from coil physical data, will be assumed as 15.95. W = assumed velocity × 60 ÷ average of specific volumes of the air as it passes through the coil. $W = (400 \times 60) \div 13.61 = 1758$ lb per hr per sq foot of coil face area. $t_1 = 83.2$ F, $t_{d1} = 63.0$ F, $t_2 = 59$ F, $t_{d2} = 56$ F. The temperatures and specific volumes were obtained from the psychrometric chart. Therefore $\frac{10.7 \times 15.95 \times N}{0.23 \times 1758} = \log_e \frac{(83.2 - 63.0)}{(59 - 56)} = \log_e \frac{20.2}{3} = \log_e 6.74 = 1.91$, and $N = 1.91 \times 0.23 \times 1758 \div 10.7 \times 15.95 = 4.51$. Four rows of tubes will be used, and equation 159 rearranged will be applied again to find the actual value of $t_2 - t_{d2}$. $\text{Log}_e \dfrac{83.2 - 63.0}{t_2 - t_{d2}} = \dfrac{10.7 \times 15.95 \times 4}{0.23 \times 1758} = 1.69$. $\dfrac{83.2 - 63.0}{t_2 - t_{d2}} = \text{antilog}_e\ 1.69 = 5.395$. $t_2 - t_{d2} = (83.2 - 63.0) \div 5.395 = 3.74$. The state point of the air leaving the coil is then at a point on the coil load-ratio line of Fig. 345 at a horizontal distance of 3.74 F from the saturation curve. The dry-bulb temperature is 59.7 F, the relative humidity 88 per cent, the dew-point temperature 56 F, and the enthalpy 24.8 Btu per lb of dry air.

The coil-surface temperature t_s may now be found by applying equation 160, $t_s = (Rt_2 - t_1) \div (R - 1)$. $R = (t_1 - t_{d1}) \div (t_2 - t_{d2}) = 20.2 \div 3.74 = 5.40$, and $t_s = (5.4 \times 59.7 - 83.2) \div (5.4 - 1) = 239.3 \div 4.40 = 54.3$ F.

The required refrigerant temperature may be found by solving for t_r in equation 161. $H_t = \dfrac{h_r}{R_s} A_s N(t_s - t_r)$. H_t, which must be calculated from available data, $= W(h_1 - h_2) = 1758(33.4 - 24.8) = 15{,}500$ Btu per hr per sq ft of coil-face area, h_1 and h_2 being the enthalpies of the air-vapor mixture entering and leaving the coil. h_r, which should be obtained from a test of a representative coil supplied with the same refrigerant, will be assumed as 325 since the refrigerant is Freon. R_s from coil physical data = 13.6 sq ft of air-side surface per sq ft of refrigerant-side surface. Therefore $15{,}500 = \dfrac{325}{13.6} \times 15.95 \times 4 \times (54.3 - t_r)$, or $(54.3 - t_r) = (15{,}500 \times 13.6) \div (325 \times 15.95 \times 4) = 10.16$ F, and $t_r = 44.14$ F.

The weight of air to be cooled and dehumidified will be considerably greater than in the preceding example because the enthalpy of the air leaving the direct-expansion coil is higher than that leaving the washer. The weight of air which must be cooled may be found as in the example of Art. 420. $W = 128{,}183 \div (31.2 - 24.8) = 20{,}050$ lb dry air per hr. The average volume of the air passing through the coil $= 20{,}050 \times 13.61 \div 60 = 4550$ cfm, and the required face area $= 4550 \div 400 = 11.38$ sq ft.

Therefore a direct-expansion coil of the type selected, four rows deep, operated with a refrigerant temperature of 44.14 F and having a face area of 11.38 sq ft could be used for cooling and dehumidifying the air in the example of the previous article. A commercially available coil would be selected and the velocity adjusted to match the exact face area.

The humidity method can be applied in the same manner to the selection of a coil in which the cooling medium is to be chilled water, provided that the applicable refrigerant-side film coefficient h_r is available.

If the required room load-ratio line does not intersect the saturation curve, the coil load-ratio line should be drawn to intersect it at the point where the other two curves are closest. The value of the term $(t_2 - t_{d2})$ is then taken as the horizontal distance between this point of intersection and the saturation curve when equation 159 is used to determine the required number of rows of tubes. If the room load-ratio line does not at any point pass close to the saturation curve, a small number of tubes and a very low refrigerant temperature will be indicated. If for any reason it is desired to use a higher refrigerant temperature than that found by the humidity method as usually applied, a coil load-ratio line having a lesser slope and which does not intersect the room load-ratio line may be drawn. The humidity method may then be applied by using the assumed coil load-ratio line, but in this case it will be necessary to reheat the air after it leaves the coil in order to move the state point horizontally to a position on the room load-ratio line. Figure 346 illustrates a case where 100 per cent outdoor air is cooled and dehumidified by a coil which is selected and operated to deliver air at 57 F and a relative humidity of 90 per cent. The air is then reheated to 67 F at which temperature the state point is on the required room load-ratio line.

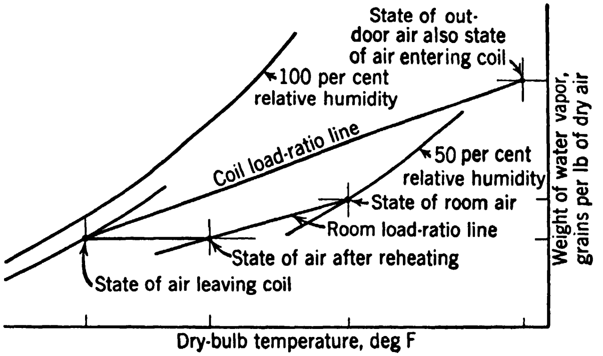

State of outdoor air also state of air entering coil
100 per cent relative humidity
Coil load-ratio line
50 per cent relative humidity
State of room air
Room load-ratio line
State of air after reheating
State of air leaving coil
Dry-bulb temperature, deg F
Weight of water vapor, grains per lb of dry air

FIG. 346. Use of a psychrometric chart in cooling-coil selection when reheating is necessary.

Where all the air passing through the coil is recirculated from the room, the state of the air entering the cooling unit is identical with that of the room air, and the two load-ratio lines will coincide unless it is necessary to use reheat. The two load-ratio lines will also coincide where the state of the room air may be adjusted so that this line can pass through it and that of the air entering the coil.

PROBLEMS

1. A theater having a seating capacity to accommodate 400 persons is to be provided with an all-year air-conditioning plant. Winter outdoor design conditions are 22 F dbt and 73 per cent relative humidity. Summer outdoor design conditions at the time of peak cooling load are 92 F dbt and 77 F wbt. The estimated heat losses from the building under winter design conditions owing to conduction through all exposed areas equal 200,000 Btu per hr. The estimated heat gain at the time of peak cooling load from all sources except occupants and fresh air equals 100,000 Btu per hr of sensible heat. Assume for design purposes that the plant will handle 100 per cent fresh air in an amount adequate to meet ventilation requirements. (Use 12 cfm per person measured under standard conditions, density 0.075 lb per cu ft.) Moisture gain from sources other than the fresh air and the occupants may be neglected. Infiltration of outdoor air may be neglected. Assume that the apparatus used will be the type that is shown in Fig. 338. The air in the conditioned space is to be maintained at 72 F and 40 per cent relative humidity during winter operation. During summer weather the inside-air conditions are to be 80 F dbt together with a relative humidity of 50 per cent. (*a*) Calculate the sensible heat factor for summer peak load conditions. (*b*) Establish the room-load-ratio line, and calculate the required temperature of the spray water for summer peak-load conditions, assuming that the air will leave the washer in a saturated state at the same temperature. Will it be necessary to use reheat? If so, specify the temperature to which the air is reheated, and calculate the heat required from the reheat coil in Btu per hr. (*c*) Calculate the refrigerating capacity which must be supplied to maintain the spray water at the proper temperature under summer peak load conditions. (*d*) Calculate the volume of the processed air and the volume of an equal weight of by-passed air. Find the temperature of the resulting mixture. (*e*) Calculate the required temperature of the spray water for maintaining the specified indoor condition during winter design conditions when the theater is unoccupied. Asume that the air will leave the washer in a saturated state at the spray water temperature. (*f*) Calculate the heat required from the preheater coil to raise the dry-bulb temperature of the incoming air to that of the spray water found in (*e*). (*g*) Calculate the heat required in Btu per hour from the water heater in maintaining the spray water temperature at the required level under the conditions of (*e*). (*h*) Calculate the dry-bulb temperature to which the air must be heated in the reheater coil to maintain the specified room temperature under the conditions of part (*e*). (*i*) Calculate the heat required from the reheater in Btu per hour under the conditions of part (*e*).

2. Assume that the theater of problem 1 is to be served by an air-conditioning plant of the type shown in Figure 337 which incorporates a direct-expansion cooling coil supplied with liquid Freon, F-12, from a remote condensing unit. Based on a face area velocity of 400 fpm and using the humidity method as illustrated in the example of Art. 421, calculate the following items for a cooling and dehumidifying coil of the type for which physical data are given in Table 130. (*a*) Will it be necessary to use reheat? If so, find the required temperature before and after reheating. (*b*) number of rows of tubes required; (*c*) required mean outside surface temperature of coil, F; (*d*) required refrigerant temperature, F; (*e*) required face area, sq ft; (*f*) refrigeration required to supply the coil, tons.

3. This is the same as problem 2 except that chilled water is to be used as the refrigerant instead of Freon, F-12. Assume that the water velocity is 200 fpm.

CHAPTER 20

AUTOMATIC CONTROLS—PRESSURE, TEMPERATURE, AND HUMIDITY

422. Necessity for Automatic Controls. Pressure-actuated damper regulators for steam boilers and thermally acting units for hot-water heaters have been discussed in Chap. 8 as a means of limiting the combustion rates of hand-fired units so that excessive pressures and temperatures do not lead to dangerous operating conditions. Another simple device for limiting the combustion rate, as shown by Fig. 347, is a thermostatically controlled damper regulator applied to the operation of the draft and check dampers of a hand-fired gravity-flow warm-air furnace. Such equipment is also applicable to hand-fired steam and hot-water boilers. As shown by Fig. 347 a room thermostat makes and breaks an electrical circuit which is energized from the electrical service of the building. The action of the thermostat is to start and to stop the damper motor.

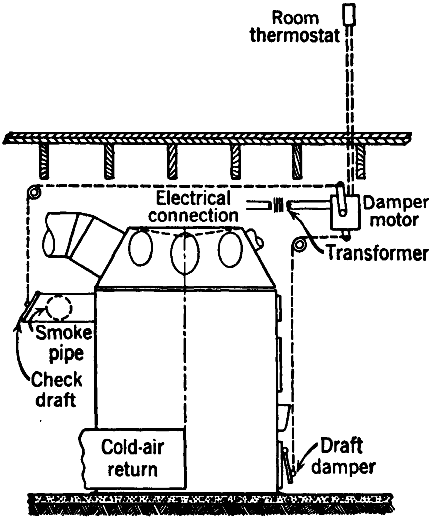


Fig. 347. Warm-air furnace with thermostat and damper motor.

All stoker-fired and oil- and gas-burning equipment must have a system of automatic controls to insure its proper functioning; the same is true in heating systems where mechanically circulated fluids are hot water and warm air. Automatic controls of various sorts, in heating and air-conditioning systems are either desirable or a necessity from the standpoint of safety, human comfort, proper functioning of the apparatus, and economy of operation.

423. Action of Controls. Generally automatic controls function either (1) positively to on or off positions; or (2) move slowly in modu-

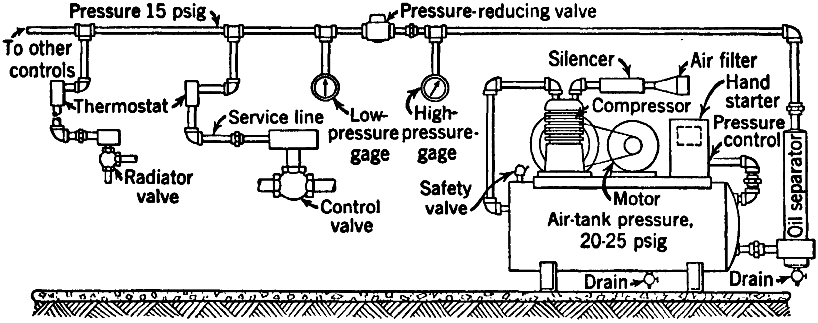


FIG. 348. Pneumatically operated temperature-control system.

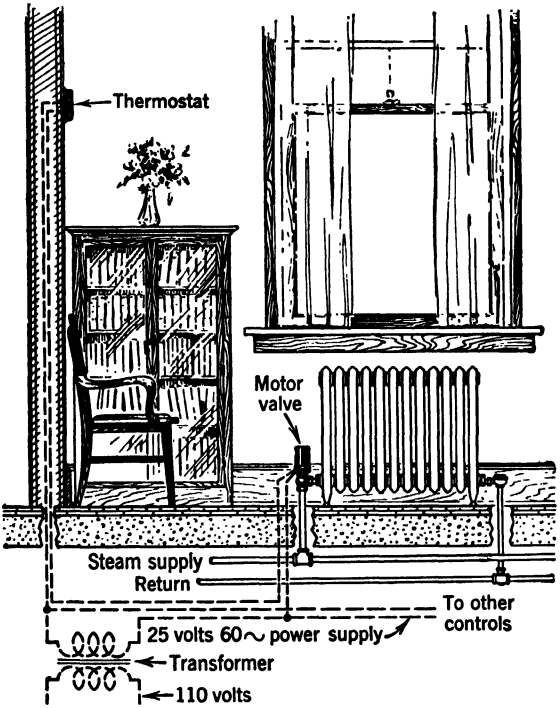
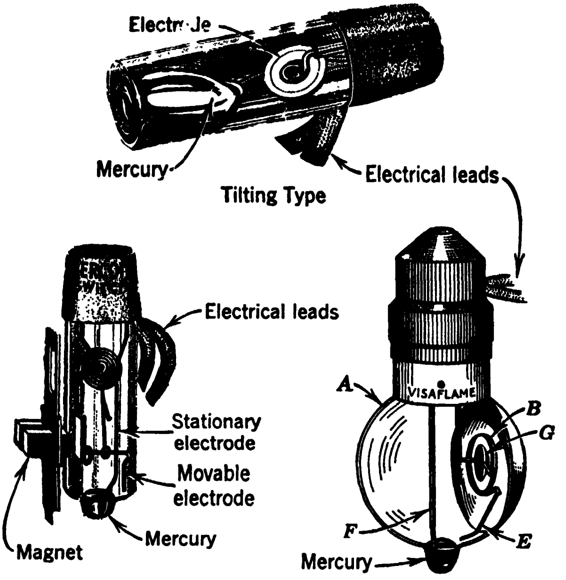


FIG. 349. Thermostatic control of a steam radiator.

lating, throttling, or proportioning actions by small increments of the total movements of valves and dampers, with the controlling device anticipating the actions produced so that over control is not effected. The motivating forces which controls employ to actuate various pieces of required apparatus are (1) those produced within the control itself, as in Fig. 120. (2) compressed air or other fluid under pressure, Fig. 348; and (3) electrical power from an external source, Fig. 349.

424. Mercury Switches. When electricity is used to operate valve and damper motors as well as other control devices, switches with non-arcing and non-corroding contacts are desirable. The three hermetically sealed glass containers, which enclose small quantities of liquid mercury and electrical terminals as indicated by Fig. 350, represent forms of switches suitable for adaptation to many heating and air-conditioning controls. These non-arcing units are divided into tilting, magnet, and light-actuated types.



Tilting Type

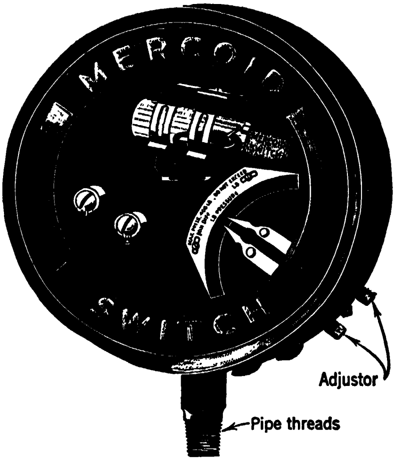
Magnet Type

Visaflame Bulb

(*The Mercoid Corp.*)

Fig. 350. Mercury switches.

The tilting type functions as the tube, when mounted on some movable part of the control, is shifted to cause the mercury within it either to cover or to uncover the internal electrodes. The tilting switch, illustrated in Fig. 350, has one electrode normally in contact with mercury and the other so placed that when mercury flows into the refractory cup beneath it an electrical circuit is completed. The magnet-type unit operates with the tube held stationary in a vertical position. One electrode dips into the mercury reservoir, and the movable electrode is brought into contact with the mercury by the action of either a permanent or an electromagnet. The action of the flat-coil spring opens the circuit when the effect of the magnet is negligible. The light-actuated switch depends upon light from some source being ultimately translated into radiant heat to warm the bimetal element *G* which shifts the movable electrode *E* into the mercury to complete an electrical circuit through electrode *F*. As shown in the illustration, light from some source, such as the flame in a furnace served by an oil burner, is transmitted through the enclosing bulb *A* to a concave reflector behind *G* which concentrates all the light waves on the bimetal coil *G*, where this opaque object causes the light waves to change to heat which affects its temperature and causes it to move an electrode and complete the circuit. Changes of ambient temperature of air and objects adjacent to the unit do not affect the operation of the contact as all parts of the unit are equally affected. If the ambient temperature is rising the coil *G* is affected and moves accordingly, but as it does so the larger and surrounding bimetal coil *B* moves in an equal and opposite direction causing a compensation to hold contact *E* in proper position.



(*The Mercoid Corp.*)

FIG. 351. Pressure control.

425. Pressure Controls. Certain pieces of refrigeration and air-conditioning equipment are operated to maintain desired pressures in condensers and duct systems. Figure 351 illustrates one form of a pressure control which functions with a tilting electrical switch actuated by a Bourdon pressure tube similar to that of the gage shown by Fig. 118. The pressure control of Fig. 351, when the pressure drops, moves the mercury switch to close an electrical circuit which in turn energizes the automatic starter of an electrical motor driving the machine to be regulated. The pressure control illustrated has two adjustments, which may be made from without, that allow it to function with a certain allowable differential of pressure near that which is to be maintained.

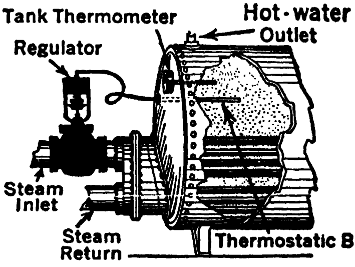
426. Thermostats. The temperature of either air or liquids is controlled by means of some device which is sensitive to temperature changes and which causes the proper valves and dampers to be moved to correct the variations. For this purpose some form of a thermostat is used which has either a volatile fluid in a flexible container or bimetal strips in straight, helical, or spiral arrangement. The strips are made of two metals, having different coefficients of linear expansion, fused together so that the length of the welded piece is equal to each of the two pieces joined. Because of the unequal expansion and contraction of the two metals, the piece thus produced bends when either warmed above or cooled below its normal temperature. The action produced by a volatile liquid in a flexible container or by a bimetal bar causes the thermostat to put other pieces of control apparatus in operation.

Thermostats are either direct or indirect in their action. When the thermostatic element is powerful enough to operate a valve or a damper without another agent as an intermediary it is direct acting. The heat-sensitive element of many thermostats is not strong enough to open and close valves and move dampers. Therefore some other medium such as compressed air, electricity, or a fluid under pressure is put into service by the thermostat to make the necessary valve and damper changes. Under such conditions the thermostat is classified as indirect as it functions to start and stop the agent producing movement in the mechanism controlling any unit.

Thermostats are further classified as positive or quick acting, graduated or modulating, and combination, which embody positive and graduated control. Positive thermostats quickly and completely open or close a valve or switch without any intermediate positions. Those which give graduated control permit slow opening and closing of valves together with intermediate positions between fully open and fully closed. Figure 349 illustrates a thermostat in connection with a direct radiator for the control of the room-air temperature.

Thermostats may be employed in securing desired psychrometric conditions of air-vapor mixtures by their use in the control of one of the following temperatures: (1) dry-bulb, (2) wet-bulb, or dew point when conditions of saturation are maintained at a predetermined value.

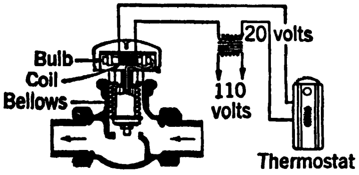
427. Hydraulic-Action Regulators. The unit of Fig. 352 consists of a thermostatic element which is placed in the tank containing a heated fluid. The bulb is filled with a volatile fluid. Pressure, created within the bulb by heat vaporizing some of the fluid, is transmitted through an armored flexible tube to the diaphragm of a valve in the steam supply line. Ordinarily the valve is held open by the



(Powers Regulator Co.)

FIG. 352. Tank regulator.

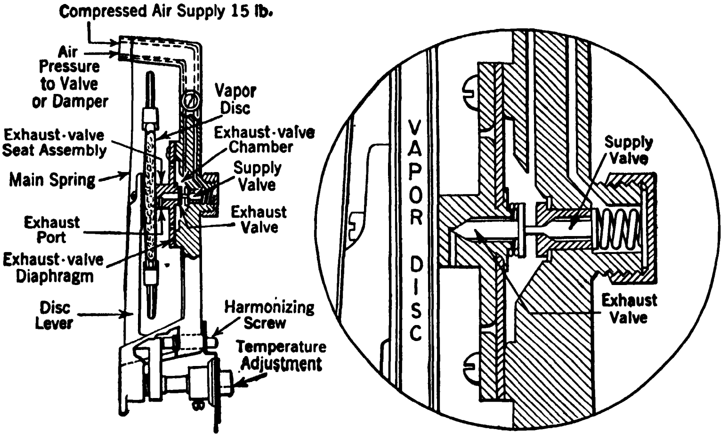
spring action of the diaphragm or springs bearing against it. The pressure developed within the bulb causes the valve to close against the resistance offered by the diaphragm, and steam is shut off from the heating coils. When the bulb temperature falls, the pressure becomes insufficient to hold the diaphragm-actuated valve closed, and the valve opens to admit steam to the heating coils.



(*The Fulton Sylphon Co.*)

FIG. 353. Hot-chamber regulator.

A thermostatically controlled globe valve is illustrated by Fig. 353 in which a sylphon bellows serves as the pressure unit. Attached to the bellows by a small tube is a bulb filled with a volatile fluid. When the thermostat is satisfied, as regards temperature, it closes an electrical circuit which includes a resistance coil wound about the liquid-filled bulb. The heating effect of the coil vaporizes some of the liquid to produce a pressure in the sylphon bellows, and the valve is closed to the passage of any material through it. Modifications of this form of control are used by various manufacturers in the production of aquastats and other temperature regulators.



(*Powers Regulator Co.*)

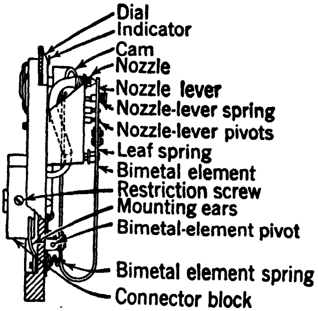
FIG. 354. Vapor-disc thermostat.

428. Room or Wall Thermostats. Figure 354 shows a room thermostat which has a corrugated hollow vapor disc or thermal element partially filled with a volatile liquid. Temperature changes cause the diaphragm to expand or contract as the pressure within it varies.

The vapor disc or diaphragm operates a double valve which controls both the supply and exhaust of compressed air for operating control valves and dampers. The thermostat of Fig. 354 is an indirect type capable of giving modulating effects.

The vapor disc and double valve are housed within a case which may or may not have a thermometer mounted upon its front. Provision is made for adjustment of the thermostat to function at various temperatures by means of a regulator which is accessible from the outside of the case. These thermostats are also made positive acting for use with one-pipe steam radiators where partial closing of the radiator valve is not permissible.

The thermostat of Fig. 355 is built to operate with compressed air as the moving force in secondary equipment such as radiator and other valves and damper motors. Air-temperature changes about the instrument are detected by a bimetal element which is bent into a U shape. The element is pivoted at a location near the bottom of the bend, and it is loaded by a spring to keep it in contact, at all times, with a cam whose position is changed as the temperature-adjustment dial is rotated. The motion of the free end of the bimetal element is transmitted through a leaf spring of the nozzle lever which rotates about pivots to either cover or to uncover the bleed port of the nozzle and thus change the amount of its opening. The spring action is sufficient to keep the nozzle opening closed against an air pressure of 15 psig in the branch line. When the nozzle port is uncovered the compressed air escapes, and the pressure in the line is below that necessary to move the secondary-control parts. When the bimetal element moves to close the nozzle port the air pressure in the secondary line increases, and the proper movement of valves and dampers is made to reduce the air temperature. These thermostats are also built with the bimetal construction reversed so that an increase of air temperature causes the element to contract and to open the nozzle air port. Decrease of air temperature produces the opposite result and the nozzle port is closed. The restriction-screw adjustment varies the amount of compressed air delivered to the thermostat and the branch line by the main-supply connection. The adjustment is made after installation to give the desired speed of operation for the devices

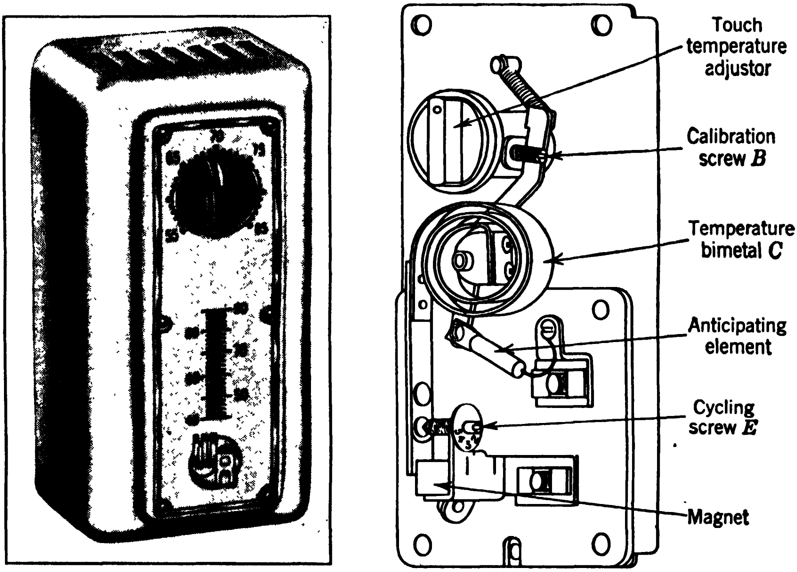


(*Minneapolis-Honeywell Regulator Co.*)

FIG. 355. Pneumatic thermostat with bimetal element.

controlled. The setting is such that the nozzle can always bleed the branch-line pressure down to one psig or less.

Bimetal thermostats, built for the actuation of secondary devices such as valves and damper motors are constructed with dual arrangements within a single case or housing. One of the elements, with a proper setting, functions to give daytime temperature control, and the other operates at night or those times when reduced air temperatures are desirable. The dual thermostats of one manufacturer are automatically changed from day service to night service, and vice versa, by varying the main air pressure from 13 to 17 psig. This action may be obtained by the manual movement of a switch or by the use of a program-operated clock. Some of the units are provided with push buttons which allow the change from night operation to day functioning when the button is moved. The push-button setting is nullified when the equipment is again indexed for day operation.



(*White-Rodgers Electric Co.*)

FIG. 356. Bimetal electric thermostat.

The construction of one form of an electrical bimetal thermostat is shown by Fig. 356 in which the temperature element *C* is mounted upon an arm pivoted at a point near to the center of the element coils. The temperature at which the thermostat functions is determined by the setting of the adjustor which in turn fixes the position of the arm on which *C* is mounted. The free end of the coiled bimetal element makes and breaks contact with the end of the cycling screw *E* which may be adjusted to give the desired frequency of the periods of on and off operation.

The electrical leads to and from the thermostat are connected by screws to its mechanism at the back of the base. When the bimetal element makes contact with screw *E* a permanent magnet located below the contact point aids in giving snap action to the movement of the free end of the bimetal element. The closure of the contacts at *E* allows electrical current to flow through a circuit which includes the heating or anticipating element, the bimetal coil *C*, and the secondary control which is to be operated. A very simple wiring connection for a single electrical thermostat is shown by Fig. 349.

The anticipating element is a resistance through which the electrical current flows and which serves as a heater unit to warm the bimetal element. The prewarming of the bimetal element causes the electrical circuit to be opened before the temperature of the air has reached that desired. Such action prevents the overrunning of the desired air temperature because of what amounts to thermal lag in the system. Similar heat-anticipating coils are used in the thermostats produced by various manufacturers.

Electrical thermostats, of the form of construction just described, are constructed with two bimetal elements one of which is used with a heater for the control of daytime temperatures and the other a low-temperature unit for night or other periods of reduced temperatures. Such an arrangement is fitted with two cycling screws and is used in connection with a timing switch (electrical clock device) which may be set to shift from one bimetal element to the other at predetermined times.

Reduction of air temperatures during definite periods of time, such as night and unoccupancy periods, may be obtained with a single bimetal thermostat operated in connection with a 24-volt electric timer clock. The temperature setting of the thermostat is not changed when a reduction of the air temperature is desired. The clock has two dials, each externally adjustable, one of which is set for the time at which the temperature is to be lowered and the other for the time at which the air temperature is to be restored to that corresponding to the setting of the thermostat. The air temperature reduction is secured by the passing of low-voltage (24-volt) current through a resistance heater located within the thermostat case. This action results from the clock closing a switch in the heater circuit. The heater action is to impart to the bimetal element an amount of heat sufficient to cause the air temperature to be lowered the desired number of degrees. When the required period of low-temperature operation has passed the clock automatically opens the heater circuit, and the thermostat maintains the temperature for which it is set. The clock unit can be set to automatically give high and low temperatures during the desired periods or either low or high temperatures continuously, depending upon the setting of a control knob located upon the front of the clock case.

429. Modulating Controls. The thermostats shown by Figs. 355 and 356 involve on-and-off operation of the equipment which they serve. Throttling, proportioning, and intermediate-acting controllers are used to give modulating effects by repositioning the moving parts of either valves, damper motors, or other controlled devices. Small changes of the total travel of the actuated parts occur when slight changes are sensed by the controllers as variations take place in the regulated conditions.

When pneumatically operated units are used throttling and modulating effects are obtained by the employment of devices in connection with the thermostats which are essentially pressure-reducing valves. These units are actuated by the controlled valves and dampers so that they reposition them to maintain the desired conditions of the controlled media. The controllers are designated as direct acting and reverse acting. A direct-acting unit increases its branch (pipe leading from the controller to the motor) air pressure on increase of the controlled condition while the reverse-acting controller increases its branch air pressure on decrease of the controlled conditions.

The pneumatic thermostat of Fig. 354 does not allow continuous loss of air from the piping by bleeding or controlled leakage. The thermostat supply and exhaust valves are not open at the same time, and both remain closed for considerable periods of time. The thermostat mechanism includes in its construction what is essentially a pressure-reducing valve. The air pressure can be held in the branch line, leading to the valve or damper motor, to give the necessary positions of the mechanism that are required to effect the desired results. This action allows the thermostat without other aids to produce modulating effects.

Modulating effects are obtained with electric thermostats by use in the circuits of what amounts to a Wheatstone bridge which when unbalanced actuates the repositioning of an element that moves a valve or a damper. Some of the details of such a device are shown by parts *a*, *b*, and *c* of Fig. 357 which indicate the construction and operation of a Minneapolis-Honeywell Regulator Company, series 90, control unit used to provide modulating or proportioning effects when either motorized valves or dampers are used. These circuits operate to position the controlled device (damper or motor valve) between fully open and completely closed conditions and thereby proportion the delivery to the need indicated by the controller mechanism. The series 90 controllers may be either (1) room thermostats, (2) insertion thermostats, (3) humidity controllers, or (4) pressure controllers.

The essential parts of the mechanism include: a transformer; a controller potentiometer; a balancing relay composed of solenoid coils C_1 and C_2, a U-shaped armature A which is pivoted at P, contacts K_1 and K_2, and a contact blade B which moves with the armature; motor coils W_1 and W_2; condenser N; the motor shaft; and the motor-balancing potentiometer. The controller and the motor potentiometers are alike electrically, and each has a 135-ohm coil of resistance wire wound on a suitable bobbin. Each potentiometer has a wiper-contact finger which may move across its resistances. A low-voltage capacitor motor drives the motor shaft through a train of speed-reducing gears, and the amount of the shaft rotation is held to 160 deg by the action of limit switches. The legs of the pivoted U-shaped armature extend into the solenoid coils C_1 and C_2, and as the armature moves about its pivot the contact arm B either may touch contact K_1, be midway between K_1 and K_2 (balanced condition), or touch K_2.

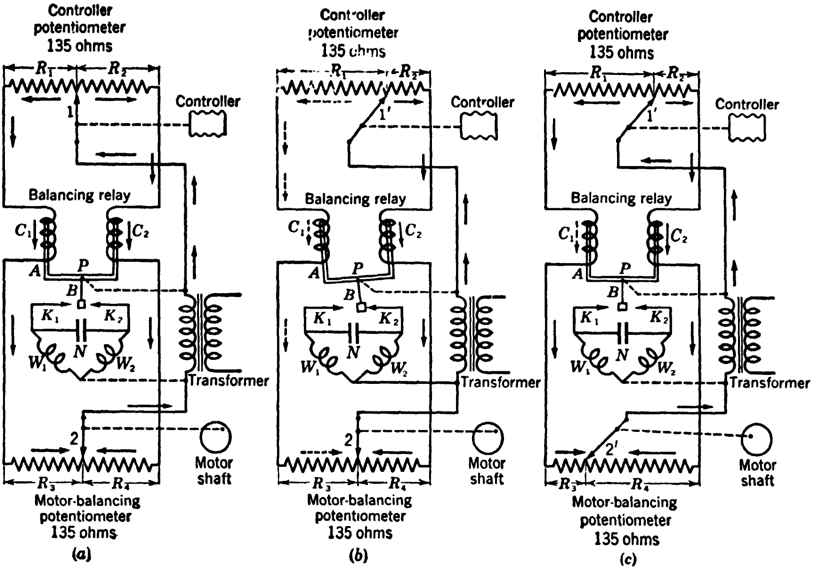
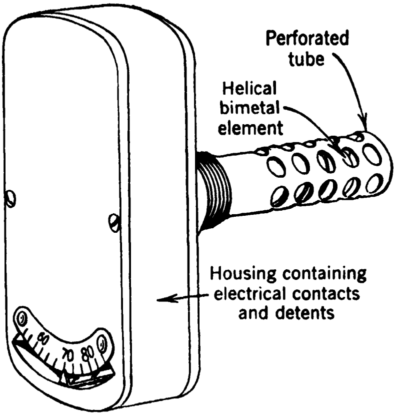


FIG. 357. Minneapolis-Honeywell, series 90, control circuit.

When the controller holds the moving contact of its potentiometer at 1, as shown by Fig. 357*a*, the motor potentiometer contact will be at 2 for a balanced condition. The currents flowing through solenoid coils C_1 and C_2 are equal, the armature A is in a position to place contact blade B midway between K_1 and K_2, and the motor is at rest. When the controller unit (thermostat or other part) causes its wiper contact to move to position 1′, Fig. 357*b*, resistance R_1 is not equal to R_2, and more current flows from the transformer through coil C_2 than flows through C_1; armature A is shifted so that its contact blade B makes an electrical circuit at K_2. The motor now receives more current through coil W_2 than it does through W_1, and it rotates in the direction corresponding to this condition. The wiper connection attached to the motor shaft then shifts the motor potentiometer contact to the position 2′ as shown by Fig. 357*c*. Resistance R_3 then becomes equal to R_2 and R_1 equals R_4. When such a condition exists the amounts of current flowing through coils W_1 and W_2 are equal, and the contact blade B is located midway between K_1 and K_2. The motor under these conditions is at rest, and the shaft has positioned the movable part of the damper or valve motor to meet the demands as indicated necessary by the controller unit. The same operating actions take place when the controller wiper contact moves over that portion of the resistance designated as R_1 and the motor wiper contact moves over R_4, except that the motor shaft rotates in an opposite direction.

430. Insertion and Immersion Thermostats. An insertion thermostat has its heat-sensitive element placed either within an air duct or a plenum chamber with the remainder of the unit mounted outside it. This arrangement is an aid in the matter of observation and adjustment of the instrument settings. The external appearance of one form of an insertion thermostat is shown by Fig. 358. This particular device has a spirally wound strip of bimetal as a thermal element which is located within a protecting shield of perforated metal. Another form of insertion thermostat uses a vapor disc, of the type shown by Fig. 354, mounted upon an



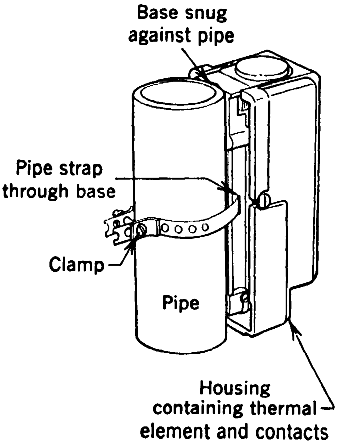
(*Barber-Colman Co.*)

FIG. 358. Insertion thermostat for a duct.

extension into a plenum chamber or duct. The remainder of the control has an external mounting. The combination of a perforated brass bulb and an invar rod (nickel steel) also serves as an actuating part in some designs.

Immersion thermostats require the use of a closed bulb to house the change of temperature-sensitive materials. Materials used are readily volatile fluids, spirally wound bimetal strips with some mercury to give good contact with the enclosing bulb, and brass bulbs enclosing invar bars. Immersion thermostats are used for the control of the temperatures of fluids such as water steam, brine, and some gaseous materials. The mounting of the mechanism, exclusive of the heat-sensitive part, is external to the fluid whose temperature is to be regulated. Those thermostats which employ bulbs filled with a volatile fluid have some of the construction characteristics indicated for hydraulic-type regulators described in Art. 427. The connection of a thermostatic bulb with the remainder of the mechanism by means of a flexible tube of any feasible length is often of considerable convenience in the location of immersion-type thermostats.

431. Surface Thermostats. Controls which function by virtue of surface-temperature changes of either pipes or ducts are often used instead of either insertion or immersion thermostats. An illustration of a snap-action electrical surface control is shown in Fig. 359 where the unit is rigidly strapped to a pipe surface so that heat may flow through its base to its thermal element.



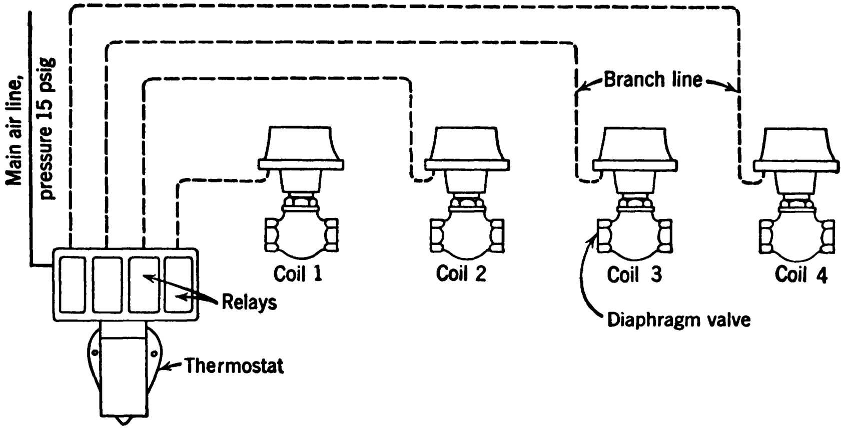
(*Perfex Corp.*)

FIG. 359. Surface thermostat.

Applications of the instrument shown by Fig. 359 are (1) in hot-water heating systems as a high-limit temperature control placed near to the boiler-water outlets; (2) as a device to prevent operation of a circulator in a hot-water heating system when the boiler-water temperature is low; and (3) when attached to the return pipe of a unit heater ahead of any valve or trap, as a control to prevent the operation of the heater fan when the heater is not receiving either steam or water heated to a sufficient temperature. This action prevents the heater from discharging inadequately warmed air into a space and thereby produce undesirable drafts.

432. Mercoid Thermostats. Non-arcing enclosed electrical switches, similar to those described in Art. 424, are often incorporated as part of thermostats employing as heat-sensitive elements, either bimetal or volatile-material-filled bulbs and bellows. These switches are operated with snap action and employ line voltages in the operation of fans, motorized valves, damper motors, and pieces of secondary equipment.

433. Compound and Multiple Insertion Thermostats. In divided or combination systems of heating and ventilation a single instrument to control the temperature of air supplied by a fan-coil heater unit and also to regulate the heat output of direct radiators is an important piece of equipment. For this purpose some form of compound thermostat is used to positively open and close radiator valves and to operate mixing valves in air ducts. These units have applications with unit ventilators when the heater-coil valve must be operated quickly and the dampers moved gradually.



(*Johnson Service Co.*)

FIG. 360. Application of a multiple-insertion thermostat.

Multiple-insertion thermostats have a single thermostatic element which actuates a number of relays. The classifications are two, three, and four points, depending upon the number of relays or automatic switches involved. The thermostatic elements are installed in either air ducts or plenum chambers.

The application of such an instrument is shown by Fig. 360, where a single perforated brass tube together with an invar rod functions by the aid of four relays to operate the steam valves of the four sections of an air heater. In air-conditioning installations the instrument may be used as either a dry-bulb, a wet-bulb, or a dew-point control. The possibility of serving several branch lines with different characteristics exists. This condition allows the satisfactory sequential operation of steam, air, refrigerant, and three-way mixing valves together with different kinds of dampers.

434. Limit Controls. These devices include equipment to control pressures through various ranges, Art. 425, and thermostatic units of the insertion, immersion, and surface types, having the general forms of the units discussed in Arts. 430 and 431.

Pressure-limiting devices find use with mechanically fired boilers, refrigeration units, and air-conditioning equipment. An aquastat is a thermostatic device used in connection with hot-water heating plants, Fig. 373*b* to control water temperatures as the fluid leaves the boiler or those of the water within the boiler at a location near to the return connections. Outlet-water temperatures are regulated by the action of either an immersion thermostat in the boiler outlet piping or by a surface unit strapped to a bare pipe surface of the same. The low-limit aquastat maintains a minimum temperature of the boiler water so that a long period of time is not required to bring the boiler to working conditions when a demand is made for heat in the spaces served by the plant. High-limit aquastats function in an electrical circuit with a room thermostat to stop or at least retard combustion within a boiler furnace by action on the operation of either automatic-fuel-burning equipment or the damper motor of a hand-fired plant. Low-limit aquastats have the opposite function as they may be used to accelerate combustion of fuels. Insertion thermostats, called limit controls, have applications in the bonnets of both gravity-flow and mechanical warm-air furnaces. In a gravity-flow furnace installation the limit control functions to prevent the bonnet-air temperature from rising above a predetermined value by the operation of a damper motor in hand-fired installations or its controlling action in the circuit of a room thermostat in mechanically fired furnaces or in the operation of a gas valve in connection with a gas burner. When a mechanical warm-air furnace is to be operated a bonnet control may have two functions, each produced by a separate thermostatic element of the device. The actions are: (1) to prevent the bonnet-air temperature from exceeding the desired and safe value, and (2) to function by means of the second thermostatic element to prevent the fan motor from starting until a suitable air-outlet temperature has been secured. This prevents the fan from circulating air of insufficient temperature to produce uncomfortable conditions within the space served. Another thermostatic device is a stack switch placed in the flue or chimney connection of an oil-fired heating unit which prevents operation of the burner when the flame is extinguished. Such operation would lead to a dangerous condition of having unburned oil collect within a furnace.

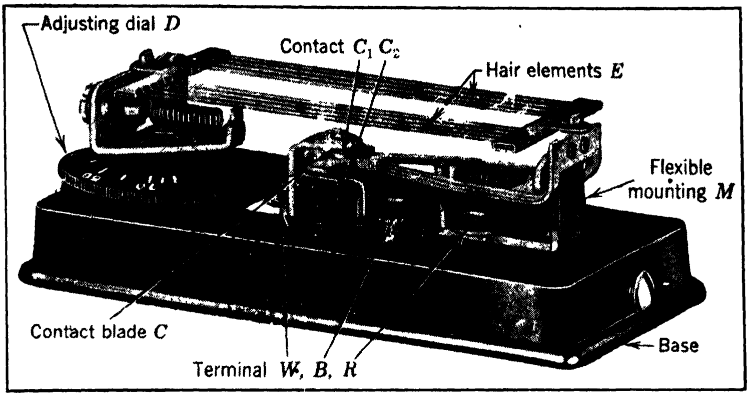
With a gas burner a thermocouple unit may be placed so that the pilot flame plays upon it to produce a small electrical current within its circuit. This action keeps a switch wired in series with those actuated by the other controls of either a boiler or a furnace closed. When the pilot flame is extinguished the thermocouple device prevents the main valve of the burner mechanism from opening and thereby prevents the accumulation of unburned gas within a furnace.

435. Humidistats. The hygrostat or the humidistat is a device which is sensitive to moisture changes in air and which will cause equipment to act to maintain a desired vapor content in the air. Often some form of a thermostat and a humidistat mechanism are placed side by side within a single housing to form a device used for the control of air conditions within a space.

The moisture-sensitive member of a humidistat can be any one of the following: a block of special wood exposing the greatest amount of end grain to the air, human hair, fiber, paper, and membranes. Other humidistats comprise a combination of wet- and dry-bulb thermostats differentially connected. One expansive element is kept moistened by means of either a wick or a water spray. The interconnection between the thermostatic elements is such that they are not affected by temperature changes alone but by changes in the relation between the wet- and dry-bulb temperature which result from a change in relative humidity. This is possible because for any definite relative humidity there is a practically straight-line relationship between the relative changes of wet- and dry-bulb temperatures.

Humidistats function to operate either humidifying or dehumidifying equipment by actuating solenoid water valves, compressed-air valves, motor-operated valves, electric-heater elements, dampers, fans, pumps, and motors. Arrangements of humidity control involve either two-position (on-and-off action) or modulating devices.

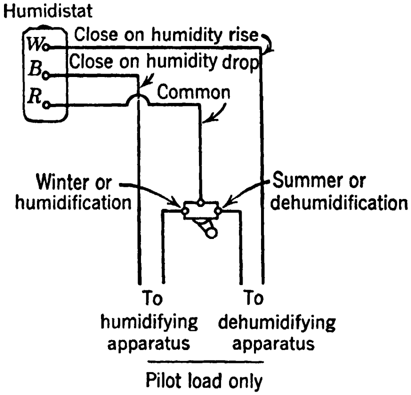
The construction of and the application of a wall-mounted human-hair element three-wire snap-acting permanent-magnet humidistat are shown by Figs. 361 and 362. The view of the humidistat mechanism shown by Fig. 361 is with its cover removed and the unit in a horizontal position. Actual installations are made with vertical placements in which the adjusting dial is at the top of the arrangement. The device includes human-hair elements E; a contact blade C; an adjusting dial D; a flexible mounting of the contact blade M; contacts C_1 and C_2; and terminal connections W, B, and R to which wires with white-, blue-, and red-colored insulations, respectively, are attached at the rear of the instrument.



(*Penn Electric Switch Co.*)

FIG. 361. Mechanism of a humidistat.

Movement of the adjusting dial D changes the tension of the hair elements. When the relative humidity of the surrounding air is above that for which the dial is set the hair elements lengthen and cause blade B to move against contact C_1 and make an electrical circuit through W and R. When the moisture of the air is below that corresponding to the dial setting the hair elements shorten in length, and blade contact is made at C_2 to complete the circuit through wires B and R. The making of the electrical circuit through wires W and R causes the equipment to function to decrease the amount of moisture being held in air; the closing of the circuit through B and R causes air humidification. In either case when conditions corresponding to the humidistat setting have been secured the proper contact is broken and the electrical circuit to the secondary control is open. Wire R is common in either circuit, and Fig. 362 shows the points of wire attachments at the back of the instrument base. Also shown in Fig. 362 is the use of a single-pole double-throw selector switch, which is

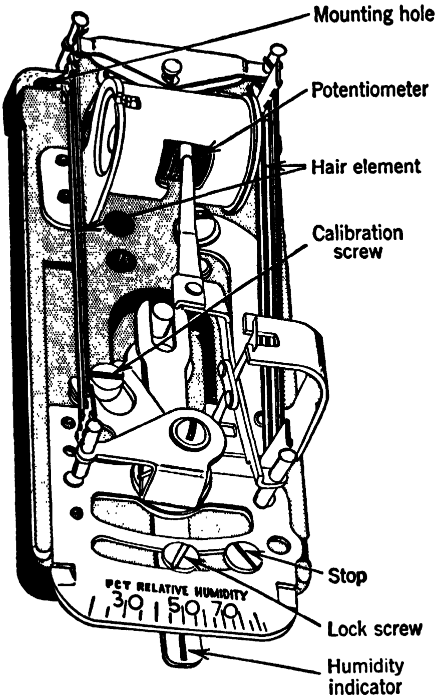


(*Penn Electric Switch Co.*)

FIG. 362. Wiring diagram for humidistat and selector switch.

manually operated, to shift the instrument control either from humidification to dehumidification apparatus, or vice versa.

Details of the construction of a human-air-element unit designed to give modulating effects are shown by Fig. 363. The functioning of the instrument depends upon the change of lengths of the human hairs with the variation of the amount of moisture in the surrounding air. The humidistat of Fig. 363 includes a potentiometer arrangement of the type discussed in Art. 429. The humidity controller, as shown in Fig. 363, is calibrated so that the sliding-wiper contact is at the mid-point of its travel over the resistance coil of the potentiometer when the room relative humidity is equal to that shown by the setting of the humidity indicator. The controller with its potentiometer, as illustrated by Fig. 363, is used in connection with a proper motor that is also fitted with a potentiometer coil. The motor operates any of the secondary-control devices that were listed earlier in this article.



(*Minneapolis-Honeywell Regulator Co.*)

FIG. 363. Internal view of humidity controller.

436. Sensitivity of Controls. In any temperature, humidity, or pressure-control system a change in the state of the conditioned material produces functioning of the apparatus. The desired corrected conditions may come from either two-position (on-and-off operation) or graduated and modulating actions. Two-position controls operate well with radiator valves installed in one-pipe steam-heating systems, on-and-off operation of stokers and oil and gas burners, pressure regulators, and various safety devices. Where high-and-low-flame oil- or gas-burner operation is required and changeable throttling actions in steam, air, gas, water, and refrigerant valves are necessary to give closer regulation, with varying load conditions, some form of equipment is necessary to give modulating effects. In order to secure modulation effects a slight change of the state of the controlled medium is necessary in order that the controlling equipment may be actuated. Under these conditions the sensitivity of the control must not be too great, otherwise the unit may be unstable and thus cause wide fluctuations above and below the desired conditions without being able to hold a steady position.

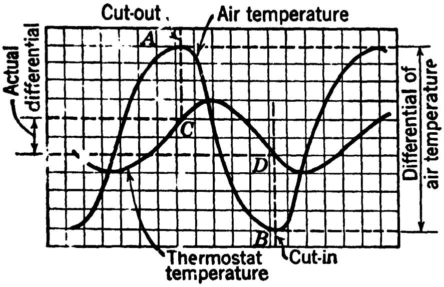


FIG. 364. Lag of air temperature.

The actions of a wall-mounting-type room thermostat are indicated by the test data[1] of Figs. 364 and 365. The data are comparative as they were taken for test conditions that may be easily duplicated and not for actual room operations which are often difficult to reproduce quickly. The air-temperature curve indicates a maximum value at the point of cut-out *A* and a minimum value at the cut-in point *B* with an air-temperature differential equal to the difference of the two. The temperature of the thermostat element fluctuated between the maximum and minimum value shown at points *C* and *D*. The actual differential of the thermostat-element temperatures is represented by its temperature difference between the points of cut-out and cut-in. A thermostat controls most accurately in air which moves rapidly and which has small changes of temperature. With the instrument operating as shown by Fig. 364, the temperature of the element lags behind that of the air with rising temperatures and remains above the temperature of the air as the value of the latter decreases.

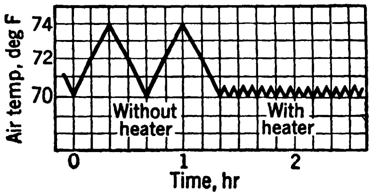


FIG. 365. Effect of compensating heater element in thermostat operation.

The anticipating or compensated form of electric thermostat, Fig. 356, employs a small heater element located within the thermostat housing. This heater begins to warm the sensitive element or blade as soon as the instrument cuts in. The action anticipates the warming of the room air and allows the thermostat to operate on smaller temperature differentials. The performances of a thermostat operating without and with a heater are shown by Fig. 365. The figure indicates that without the heat-anticipating device the maximum air-temperature differential was 4 F, and with it in service the differential was reduced to $\frac{1}{2}$ F as the unit cycled more frequently than before. Air velocity, within a thermostat, has a marked effect on the air-temperature differential when a heater is not used as a part of the apparatus construction; when the heat-sensitive part of the control is warmed by the action of an auxiliary device the effects of air velocity are largely eliminated. The sensitivity of all thermostats should be changeable by means of an adjusting screw, and also, in order to satisfy various installations, provision should be made for varying the amount of heat imparted to the thermostatic element.

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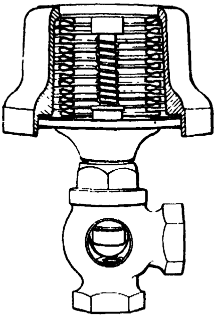
[1] "Development of Testing Apparatus for Thermostats," by D. D. Wile. *ASHVE Trans.*, Vol. 42, 1936.

437. Locations of Thermostats. The preferable location of a room thermostat is on an inside wall about 30 in. above the floor; a thermostat so placed will give better control of the air temperature in the occupied zone than one located at the 5-ft level. When a single thermostat is used to control the temperature of a structure or portions of the structure careful attention must be given to the location of the room in which it is placed. This consideration must take into account the amounts of solar radiation passing into the rooms involved, the heat liberated within the rooms controlled, and the effects of radiant heat from direct radiators upon the room occupants.

438. Air Compressors. Air for operating pneumatic controls is ordinarily at a pressure of 13 to 17 psig. When air at a higher pressure is available the necessary pressure can be secured by means of a pressure-reducing valve. Where compressed air is not available provision is made for it by the installation of small motor-driven or hydraulically operated air compressors. Electrically driven air compressors such as those in Fig. 348 are most common. The air must be free from water, oil, and dirt as the port passages in many thermostats and controllers are very small and are likely to become clogged if the air is not clean.

439. Voltage for Electrical Thermostats. The motors and controllers in many systems having electrically operated thermostatic control function with the current supplied at 20 to 25 volts. With alternating current, this low voltage is readily obtainable by stepping down the house voltage by a small transformer. If direct current is available resistances such as lamp banks are necessary to reduce the voltage. Storage batteries operating at 6 volts may also be used with direct-current controllers. Thermostats and other equipment which function at line voltages which are 115 and 230 are also used.

440. Diaphragm and Motor Valves. The pressure of a fluid such as either compressed air or a volatile liquid actuates diaphragm valves of the type shown by Fig. 366. The valve may be made to open and close quickly, or it may be only partially opened or closed, depending upon the thermostat used, i. e., positive or graduated control.

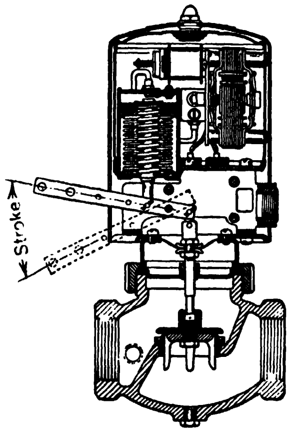


(*Johnson Service Co.*)

FIG. 366. Sylphon-bellows radiator valve.

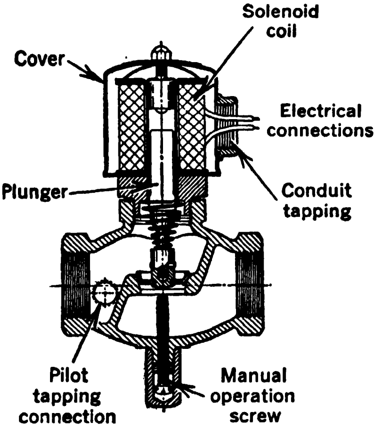
The radiator valve of Fig. 366 has a sylphon bellows of metal which together with a central coil spring holds the valve open normally. When the thermostat functions to reduce the room-air temperature compressed air is admitted, under a pressure not to exceed 15 psig, inside the casing surrounding the bellows. The external pressure on the bellows forces it to contract, thereby pushing the valve stem downward to close the valve.

In temperature-control systems using electrical power to operate the central valves a motor valve is necessary. Figure 367 shows a globe gas valve with its motor and bellows which operates an air-damper lever of a gas burner and a valve stem with a disc for the control of its gas supply. The valve opens during a period of 8 to 10 sec and closes in 2 to 3 sec. This slow motion is an aid in preventing furnace noises as gas ignites or as the flame is extinguished.



(*General Controls Co.*)

FIG. 367. Hydramotor gas valve.

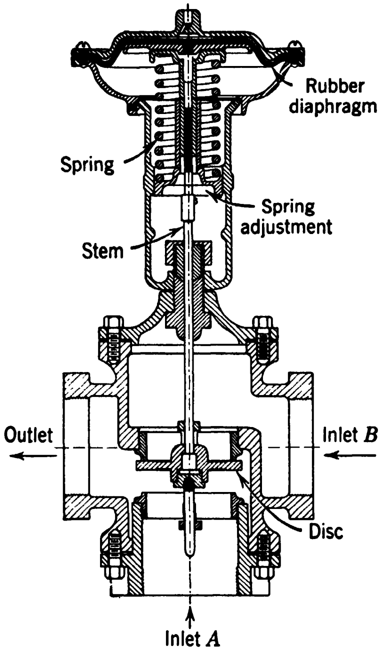


(*General Controls Co.*)

FIG. 368. Solenoid globe-type control valve.

The magnetic control valve as shown by Fig. 368 is of the electric-solenoid type and is suitable for use with gas furnaces, gas-fired steam and hot-water boilers, conversion burners, and industrial furnaces. The component parts of the valve are as indicated by Fig. 368. The valve operates under the action of either a room thermostat, limit control, or other device, which closes a circuit to energize the coil and cause its plunger to be drawn upward, by the action of the electromagnet thus formed, and thereby open the valve. In case of current failure the valve automatically closes, and the gas pressure in the supply line acts to hold the disc upon its seat. In the case of current failure the manual-opening device at the bottom of the body may be used to lift the valve disc from its seat to allow gas to pass to the burner. When the power service is renewed it is necessary to lower the operating screw to a position where automatic control under thermostatic action may occur. These valves are built for single-phase alternating current having operating potentials of 24, 115, and 230 volts and frequencies of 25 and 60 cycles.

When either heated or refrigerated water is to be mixed with other water in the operation of air washers and coil equipment, the three-way mixing valve as typified by the unit shown by Fig. 369 is useful.

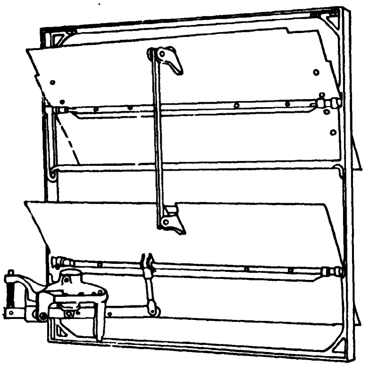


(*Johnson Service Co.*)

FIG. 369. Three-way pneumatically operated mixing valve.

Valves of the construction of Fig. 366 and which stand open, when the pressure of compressed air is not exerted upon their bellows, are classed as direct acting; when the air pressure must be applied to the bellows to open the valve the term reverse acting is used.

441. Damper Controllers. These devices are operated either electrically or by the pressure of some fluid such as compressed air. Figure 370 illustrates the application of a pneumatic controller to a two-blade mixing damper assembly. Dampers can be so arranged that a single motor will operate any number of leaves. The damper action may be either two-position (with complete opening and closing only) or modulating. Modulating may involve either gradual action, partial opening and closing, or fully open or fully closed positions. A damper controller in the form of a static-pressure regulator is shown by Fig. 260.

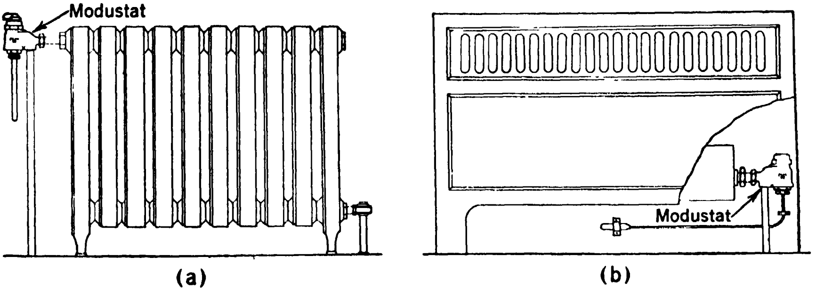


(*Johnson Service Co.*)

Fig. 370. Two-blade damper with pneumatic regulator.

442. Combination Radiator Valves and Thermostats. All two-pipe steam, vapor-, and vacuum-heating systems may be installed with a special thermostatic control at each direct radiator and convector. Such devices for regulating the room-air temperature embody a radiator valve, at the heating-unit inlet, which has a thermostatic element as an integral part.

Applications of a Minneapolis-Honeywell Modustat are indicated by Fig. 371*a* for a direct radiator and by Fig. 371*b* for a convector. The valve body, Fig. 371*a*, has a rigid extension from its lower side which is divided into two chambers by a baffle containing a small hole. A small amount of a volatile liquid is held in the bottom of the extension. Changes of air temperature around the lower part of the extension produce changes of pressure in the extension. As the liquid is gasified it passes through the small opening of the diaphragm to the upper chamber of the extension. The amount of the opening of the orifice in the valve body, through which steam passes, is determined by the gas pressure exerted upon a metal valve-actuating bellows at the bottom of the valve body. The valve is adjustable for full automatic temperature control, and it may also be used for hand control if so



(a) (b)

(*Minneapolis-Honeywell Regulator Co.*)

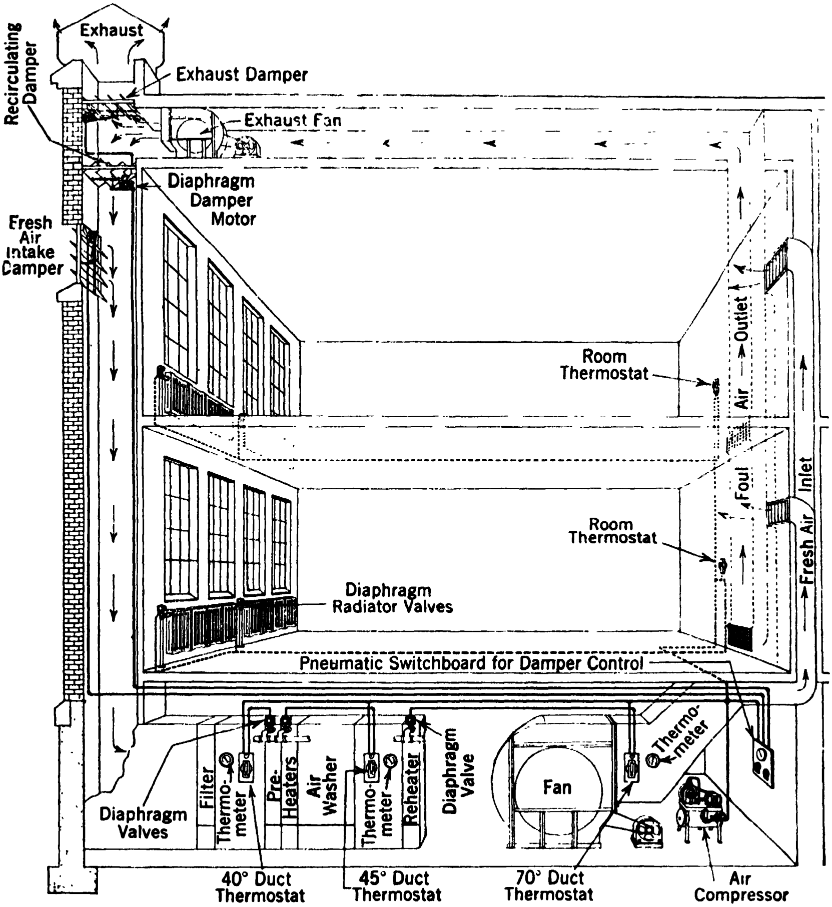
Fig. 371. Modustat temperature control.

desired. In the valve of Fig. 371*b* the rigid extension is replaced by a flexible hollow tube and a bulb containing a volatile material. The bulb is placed in any position where the temperature of the air most nearly represents that of the air returning to either the radiator or the convector to be controlled.

443. Applications of Automatic-Control Devices. The primary functions of automatic-control devices, used in heating and either comfort or industrial (winter, summer, and all-year) air-conditioning plants, are to provide the desired, the necessary, and satisfactory conditions along with economy of operation in the consumption of fuel, steam, water, power, etc. The requirements may involve only the control of air temperature; they may also include regulation of the air humidity along with its movement and distribution. Agents used for the purposes mentioned include thermostats of various types and actions, humidistats, damper and valve motors, and pressure regulators with varied applications.

Thermostats (1) control the outputs of steam radiators, unit heaters, ventilators, and air conditioners, central-fan heater plants, Fig. 372, and the circulators of hot-water heating systems as well as the fans of mechanical warm-air furnaces; and (2) regulate the burning of fuels in steam and hot-water boilers, gravity-flow and forced-circulation warm-air furnace systems, and direct-fired unit heaters. The thermostats may be set either manually or by clock-actuated switch mechanisms to give different day and night conditions, Art. 428, and intermittent operation. Small buildings may have only one thermostat for their air-temperature control; larger structures may have a thermostat in each room. The operating results obtained by each scheme are dependent to a great extent upon local conditions.

An indication of how temperature-regulating devices may be used in winter air conditioning is shown by the details of Fig. 372, which is for a split system. The unit shown permits the cleaning of a mixture of recirculated and outside air by means of an air filter prior to its passing to the preheater coils which are under the control of thermostats and pneumatically operated steam valves. The initial dry-bulb temperature of the air entering the washer is adjusted to that which is necessary to give the desired dew point when the equipment operates with unheated spray water to produce partial adiabatic saturation of the air. The final dry-bulb temperature of the air as it leaves the reheater sections (through a connecting duct section not shown) and enters the fan is fixed by adjustment of the thermostat, at the fan outlet, which controls the pneumatically operated steam valve of the reheater section. The setup of Fig. 372 is not arranged for air cooling with dehumidification. The addition of refrigeration equipment together with a three-way mixing valve, suitable piping, and additional thermostatic controls will permit the washer to serve for all-year air conditioning. Other applications of pneumatic-control equipment are shown by Figs. 348 and 360.

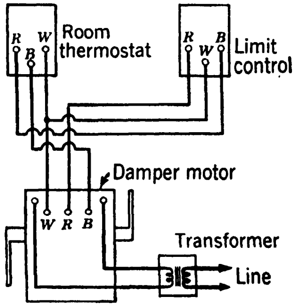


(*Powers Regulator Co.*)

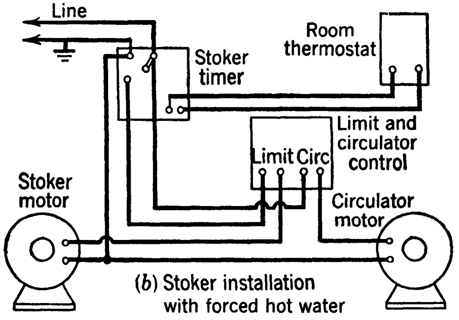
FIG. 372. Split system of heating and ventilation.

Assemblages of important parts, together with their connecting wiring, are shown by the examples of Fig. 373 for both hand- and automatically fired fuel-burning equipment and a simple unit heater.

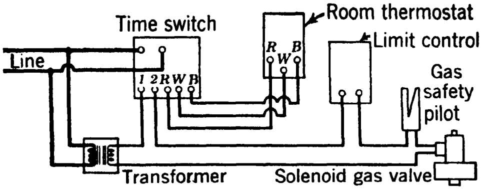
The unit-heater fan, Fig. 373*d*, is under the regulation of a room thermostat and a limit control which is some form of an aquastat. The latter prevents fan operation when the temperature of water (either condensed steam or other cooled water) leaving the heater is low. Such restriction of the fan rotation prevents cool air from being discharged into a space when its room thermostat makes a demand for heat. Other modifications of the control arrangement include placing a thermostatically operated valve in the heating-medium supply line.



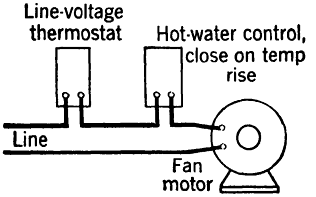
(a) Damper motor installation



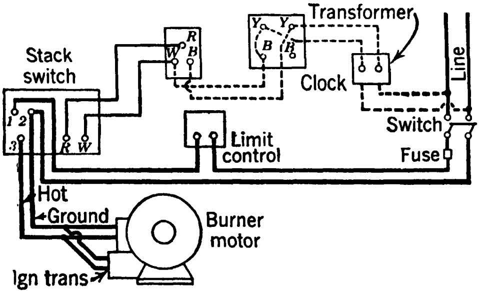
(b) Stoker installation with forced hot water



(c) Night setback circuit using solenoid gas valve



(d) Light-duty line-voltage thermostat with unit heater



(e) Oil burner for steam, hot-water, or gravity warm-air system

——— Low voltage
——— High voltage
------ Clock circuit
Letters *B*, *R*, *W*, and *Y* refer to wiring color code

(*White-Rodgers Electric Co. and Penn Electric Switch Co.*)

FIG. 373. Typical wiring connections for automatic controls.

Examples *a*, *b*, *c*, and *e* of Fig. 373 have some form of limit control as a safety feature. With steam boilers the device is usually a pressure-actuated switch; hot-water boilers have the maximum outlet-water temperature regulated by an aquastat (either a surface or an immersion thermostat); and in warm-air furnaces the maximum temperatures of the air leaving their casing plenums are held to reasonable values by insertion thermostats placed in the air stream. The oil burner has a protective device, the stack switch, which prevents fuel from being fed to a furnace without being burned. The safety pilot control, included in example *c* of Fig. 373, functions to prevent the solenoid valve from passing gas to the main burner when the lighter flame is extinguished. The stoker installation, case *b* of Fig. 373, includes equipment to cause fuel to be fed to the furnace, for short intervals, irrespective of the demands for heat in order that the fire will not burn out. The illustrations of the oil-burner and the gas-burner hookups include arrangements whereby setback of air temperatures may be had at desired intervals; this feature is possible with all the examples shown, provided that the proper equipment is included. Case *b*, of Fig. 373, provides for the operation of a water circulator independently of the stoker. When mechanical warm-air furnaces are operated a second thermal element of the bonnet-limit controls prevents operation of the system fan until the outlet-air temperature is equal to or above a certain prescribed minimum value. This feature prevents the fan from discharging air of insufficient temperature into the spaces which are to be heated.

A summation of prior statements leads to the following. (1) All steam boilers require some form of pressure-actuated control (see Figs. 120 and 373). (2) All hot-water boilers need some form of an aquastat (see Figs. 120 and 373) to control the outlet water temperature. (3) Stokers need a hold-fire device, gas burners require a safety pilot, and oil burners necessitate a stack switch or other device to prevent operation when combustion is not taking place. The items are in addition to the requirements given by statements 1 and 2. (4) Gravity-flow warm-air furnaces should have a bonnet control; all mechanical installations require a limit control to prevent excessive air temperatures and to provide for a minimum air temperature below which the system air blower cannot be started. Forced-circulation hot-water systems require the automatic operation of the circulator.

Thermostatic elements incorporated as parts of radiator supply valves have applications as explained in Art. 442. The balancing of and the regulation of the heat outputs of steam radiators, with resultant fuel savings, may be obtained by the use of orifices as indicated in Art. 229. Balancing of hot-water heating systems and regulation of their radiators are also effected by the use of either orifices or restrictor fittings placed at proper locations.

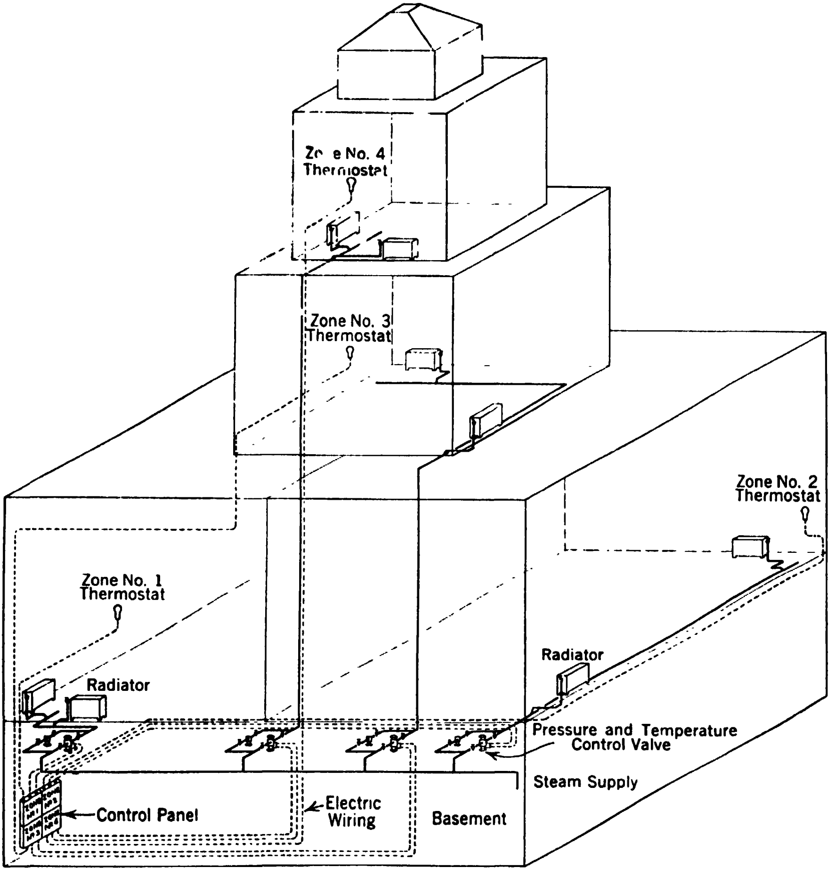
Satisfactory air-temperature control within a structure is often accomplished by dividing the building into zones wherein various devices function in each section independently of those in other spaces. Dual control of heating systems may be had by the use of both indoor and outdoor thermostatic elements so placed that the output of a plant or sections of it may be modified as weather conditions change.

444. Zone Control. Automatic regulation of the air temperature in each space division of a building may require considerable equipment with its attendant initial costs. Satisfactory air-temperature control in all the spaces of a building by means of a single thermostat, located in a representative key room, is almost impossible of attainment. The situation exists because of the nature of the building occupancy; the varied air-temperature requirements of the occupants; the building construction; exposures of the sides of the structure; wind velocities and directions; outside-air temperatures; and the varying effects of solar radiation depending upon the season of the year, the time of the day, and atmospheric conditions.

Therefore, for the purposes of air-temperature regulation, it becomes desirable to divide buildings into zones, each of which is separately controlled and is provided with the necessary arrangements of piping and duct work for any steam, hot-water, panel-heating, or forced-air-circulation heating system. Although zone control is usually given most attention in multistory buildings and industrial structures spread over considerable ground areas, it is sometimes possible to secure beneficial results in comparatively small residences by its use. The possibilities of zoning a multistory building are indicated by the example of Fig. 374; here space divisions are made along both horizontal and vertical planes, and the steam pressures maintained in the piping of the various sections are under control of thermostats serving them.

445. Combinations of Indoor and Outdoor Thermostats. All such arrangements have an outdoor thermostatic element which is properly placed and shielded from the sun. This element may be either a resistance coil of fine wire, a bimetal part in combination with a resistance coil, or a liquid-filled bulb connected by a small flexible tube to the system. The outside element is affected by the temperature of the air surrounding it, by wind, and by other weather conditions. In a wire coil a change of temperature affects its electrical resistance and the amount of current flowing in its circuit; the liquid-filled bulbs have expansion or contraction of their fluid as the conditions of their exposure vary.

The essential parts of a system of inside-outside thermostatic control is shown, in connection with a differential-vacuum heating system, by Fig. 164. The outside resistance coil is located within the window-mounted automatic selector and functions to indicate the demand for heat. Two resistance coils are placed within the heat balancer; one above and one below its air-heating surface. They measure the average temperature of the air as it enters and leaves the balancer.



(*C. A. Dunham Co.*)

FIG. 374. Zone temperature control.

The average air-temperature difference is a measure of the heat supply. The room resistance thermometer is a limit control the action of which is explained later.

The actions of the resistance coils are transmitted by the amount of current flowing through them to the main control panel which contains Wheatstone-bridge circuits in combination with a galvanometer and a relay. The galvanometer and the relay combination (galvarelay) are separately housed within the control panel. When the master switch is at its control station a feeler bar of the galvarelay is operated by a solenoid and a synchronous motor to determine at frequent intervals whether the feeler is at its central position or is deflected to one side or the other. Deflection to the right indicates a demand for heat, and a relay is closed to cause the main-control valve in the steam supply line to be opened a small amount. Deflection to the left occurs when the heat supply is above the demands, and a relay is closed to effect a closure of the control valve. When the supply of and the demand for heat are balanced no relay contact is made and the position of the control valve remains unchanged. Thus the galvanometer determines the balance between the heat supply and the demands for it and through the relay governs the opening or closing of the steam valve by small increments. The room resistance thermometer checks the demand for heat against the supply provided by the action of the steam valve. When the measured temperature is high the relay closes the valve only; it cannot open it. If the room temperature is low the room resistance thermometer acting through the remainder of the control causes the relay to open the steam valve; it cannot close it. Thus the room resistance thermometer serves to limit the actions of the selector and the heat balancer.

Depending upon the demands for heat the control valve may be fully open, fully closed, or in some intermediate position of opening. The differential controller is used to start and stop the vacuum pump, and its operation is governed by the pressure differential between the steam-supply main and the return piping. When a building is zoned one such control system is required for each section of the structure.

The Johnson Electronic Duo-Stat functions in somewhat similar manner as the Dunham control. Coils of resistance wire placed in sealed tubes are used as the heat-sensitive elements. The outdoor coil, properly shielded from the sun, is located close to a wall surface. Inside the building resistance coils ranging from 1 to 4 in number may be used. With steam and forced-circulation hot-water radiators one heat-disseminator unit has three or four resistance coils clamped firmly, at different locations, to its exposed surfaces. With gravity-flow hot-water plants one bulb is fastened firmly against a bare surface of a pipe in a supply main. Furnace systems have a bulb location within a supply duct conveying warmed air. All resistance coils are connected in series, and as their temperatures vary changes of voltage intensity are transmitted to the main controller. The controller is an electronic device having two duo-triode vacuum tubes for amplification purposes and also a Wheatstone bridge. The series-connected coils form one arm of the Wheatstone bridge. When the bridge is out of balance, because of cooling of the thermostatic elements, a small alternating signal voltage flows in the bridge galvanometer circuit and thence to three stages of amplification produced by the two vacuum tubes. The first two stages of amplification are provided by one tube and the third by the second tube. The first three plates of the two tubes are supplied with direct-current potential by the remaining plate of the second tube which serves as a half-wave rectifier. A special built-in transformer provides alternating-current power for the amplifier and the bridge circuit. Cooling of the outside coil reduces its resistance, and the bridge is thrown out of balance. This condition causes a small signal voltage to flow to the tubes where it is amplified sufficiently to operate the control relay and thereby operate a device to supply heat. Heating the indoor resistance coils increases their opposition to any current flow, and the bridge is brought into balance. With this condition the amplifier receives no signal voltage; thus the control relay moves a controlling device to turn off the heat supply. With this scheme of operation the temperature at which the heating plant is to operate is determined by the outdoor temperature. This particular device may operate the valve in the steam-supply line serving radiators fitted with inlet orifices, and it may operate automatic fuel-burning devices, water circulators, and blowers.

The Electric Duo-Stat, without electronic equipment, actuates electrically operated valves to give satisfactory control but does not have the flexibility that may be secured with the apparatus just described. The Pneumatic Duo-Stat is equipped with thermostatic bulbs with flexible connecting tubing. Bulb locations may be as given for resistance coils. The action is to operate a pneumatic relay which varies the pressure of compressed air in the branch lines serving controlled devices; the controlled valves and other appliances of equipment respond to the demands made.

Webster Moderator E Series controls employ a shielded outside thermostat made up of a flat coil of bimetal material and a wound resistance over which the element moves a contact arm. The EH-10 series of controls have liquid-filled outdoor thermostatic bulbs in connection with small flexible connecting tubing. Neither the E nor the EH Series units have an indoor thermostatic element. The following descriptions are pertinent to the E-5 and E-6 units.

Essential parts of the Moderator System are the outside thermostat, the variator, the control cabinet, the main steam-supply control valve, an orifice in each steam-radiator inlet (sometimes the orifice placement may be in a steam main), and a transformer to step the electrical service potential down from 115 to 25 volts for use in the control panel and with the motorized valves, etc. The first three parts listed work together to position the control valve to give the proper amount of steam required for the building heating.

The outside thermostat is enclosed within a metal casing which is further housed within a sun shield. This arrangement is made so that air may circulate between the inner casing and the sun shield. One end of the flat bimetal coil is rigidly fastened; its other end is attached to a shaft which it rotates as temperature changes occur. The shaft has a contact arm attached to it which is moved over the wound wire coil to vary its effective resistance as air-temperature changes occur. The contact arm has a pointer which indicates, on a scale graduated in degrees, the existing outside-air temperature. With the control switch closed, the variator position at normal, and the contact-arm position at 70 F or higher the steam-supply control valve is closed.

The variator is used as a means of hand modification of the amount of steam admitted to the heating system. With the E-5 control the variator is mounted within a separate cabinet placed at any convenient location; with the E-6 installations the variator is placed on the door of the main-control cabinet. The variator has a variable resistor. An external knob placed on the end of a shaft, which carries a contact arm, permits rotation of the shaft through an arc of 180 deg to change the resistance of the circuit in which it is placed. The resistor is connected to a terminal board by three wires. A three-position switch mounted above the knob may be used to give the following operating conditions: (1) no heat with the steam-supply valve closed, (2) full automatic heating with the outside thermostat in operation, and (3) full heat which gives opening of the steam valve to provide the full amount of steam for the base outside temperature when the variator is set at normal. These setting positions permit (1) shutoff of steam at night or other periods, (2) full automatic operation of the plant, or (3) extra steam or heat during warm-up periods.

The control cabinet houses an insulated panel upon which are mounted a special U tube containing mercury and oil and a subpanel bearing an electronic-control unit. Included also are terminals, jumpers, and wiring. The U tube embodies a vertical transparent tube which contains an electrical resistance of wire wound upon a rod of insulating material. Connection from a steam-supply main is made to a mercury reservoir which is further connected to the bottom of the transparent tube. At the top of the transparent tube is a reservoir containing oil. A connection with this well is made to a return main when such is a part of the heating system; with open heating systems the oil well is freely vented to the atmosphere. The function of the oil, which rests on the top of mercury within the tube, is to keep the mercury and the transparent tube clean; the oil well serves as a storage space for its fluid. Variations of the steam-pressure differential cause the mercury to rise and fall within the transparent tube and thus change the effective resistance of its wound coil. The electronic control unit has a vacuum-tube amplifier, two relays, and three legs of a bridge circuit. Energizing power is received from the transformer. The resistance of the outdoor thermostat, the variable resistance of the transparent tube, and the variator resistance are connected in series to form the fourth leg of the bridge. A correct total resistance in the fourth leg of the bridge causes it to be balanced with no voltage applied to the input of the electronic amplifier; consequently neither amplifier output relay is energized, and the contacts stand at neutral with no current flowing in the motor being controlled. A change of temperature conditions will cause the bridge to become unbalanced; thereupon one or the other of the relays will become energized, and the control functions to correct conditions.

With an outside temperature of 70 F, the thermostat resistance is zero, the U-tube resistance maximum, the bridge balanced with the right-hand relay energized, the steam valve closed, and the motor is idle. A drop of the outdoor temperature causes the arm of the thermostat to add resistance and unbalance the bridge to de-energize the right-hand relay and to energize the left-hand relay so that the motor opens the steam valve. As a result of pressure-differential increase in the system, mercury rising in the transparent tube cuts out resistance until the bridge goes through balance to unbalance in the opposite direction. The results are: the right-hand relay is energized, the left-hand relay is de-energized, the motor is reversed to close the valve, and the pressure differential is decreased. As the mercury column begins to fall the system again passes through balance. The motor reversal is repeated as the system seeks for a balance. Oscillation stops when the correct pressure differential is secured to cause the proper amount of heat to be supplied to care for increased demands; when this occurs the valve position and the pressure differential remain unchanged until variations of outside-air temperature occasion different demands for heat.

446. Savings Produced by Air-Temperature Regulation. The effects of the installation and operation of control systems together with those resulting from any attendant changes in the equipment of existing buildings are given by the data[2] of Table 131. Several fac-

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[2] "Heat Requirements of Buildings," by J. H. Walker and G. H. Tuttle, *ASHVE Trans.*, Vol. 41, 1935.

TABLE 131

BUILDING STEAM CONSUMPTION WITH AND WITHOUT AIR-TEMPERATURE CONTROL EQUIPMENT

Type of Control	Number of Buildings	Steam For Heating, Lb per Deg Day		Savings, Per Cent
		Before	After	
1. Subatmospheric pressure system	5	1720	1450	15.7
2. Orifice system controlled by an outdoor thermostat	2	2039	1600	21.6
3. Orifice system controlled by a group of outdoor thermostats	1	4580	4255	12.2
4. Orifice system, manual control	3	7930	6750	14.9
5. Orifice system, electric zone control	2	4110	4025	2.1
6. Intermittent control, electric timer	5	1340	1200	10.4
7. Intermittent control, actuated by outdoor and radiator temperatures	14	772	623	19.3
8. Central thermostat	19	776	585	24.6

tors may have produced the reported savings, but the results indicate in a comparative way what may be accomplished.

The fuel savings that may be secured by reductions of air temperatures by either manually or clock-adjusted thermostats are dependent to a large extent upon local conditions. Involved are the size of the structure, its construction, exposures, and installed equipment. Residences provided with well-insulated walls and ceilings below attic spaces and fitted with either weather stripping or with storm sash and doors, to reduce glass and infiltration losses, will show lower savings in fuel consumption than those structures having poor-wall construction, large single-thickness glass areas, with heavy infiltration and exfiltration of air about windows and doors as well as through other cracks. The amount of air-temperature reduction involved, the period of reduced air temperatures, the warm-up period permissible, the nature of the space occupancy, the type of the heating plant and its installed equipment, and the fuel used as well as its method of burning have an influence on the results obtained.

Test results have been obtained, at the well-insulated weather-stripped I = B = R Research Home at the University of Illinois, relative to the effects of operation with reduced-air temperatures, throughout periods of thermostat night-setback during the heating season. The Research Home tests involved the use of an oil-fired, forced-circulation, one-main, hot-water heating system. The tests[3] were made with the heating plant operating with and without a flow-control valve near the boiler-water outlet and a low-limit aquastat placed in the rear section of the cast-iron boiler about 26 in. above the bottoms of the water legs.

When the heating system was fitted with a flow-control valve and an aquastat to maintain a minimum boiler-water temperature of 165 F, the thermostat was set at 66 F at 10:30 P.M. and reset at 5:30 A.M. each day. A saving of approximately 10 per cent was secured in the seasonal fuel consumption and burner operating time together with a reduction of the operating time of the circulator of approximately 5.5 per cent. Without the flow-control valve and the low-limit aquastat only slight savings in the burner and circulator operating times and in the consumption of fuel per heating season were secured. The latter method of plant operation required longer warm-up periods during the morning than when a flow control valve and an aquastat were in use. The added losses during the longer warm-up periods tended to offset savings due to the night setbacks. Night reduction of inside-air temperatures had no effect on temperature conditions in the house during the day exclusive of those during the morning warm-ups when over-runs of from 2 to 5 F were observed at the ceiling level with none at the 30-in level. During average winter weather the maximum drop of indoor temperature was 6 F during the 7.5-hr period of night setback of the thermostat.

The curves of Fig. 375 give the steam consumption of an office building, supplied with steam from the mains of a district-heating utility, during two 24-hr periods. The conditions of operation were without and with reduction of the building air temperatures during a night period by shutoff of the building steam supply. The steam-radiator surface of the structure was sufficient to bring the building air temperatures to a satisfactory level within a reasonable period of time after the use of steam was resumed. Identical operating conditions are difficult to obtain during two 24-hr periods of building heating. According to the data of Fig. 375 the chief difference in the operating conditions, during the two 24-hr periods, was the direction of the wind and its mean velocity. The time interval during which the steam was shut off was somewhat favored with a lower average wind velocity. The saving in the consumption of steam, by the shutting off of the steam supply, was [(29,170 − 15,000)100]/29,170 = 48.5 per cent, which is for one day and does not represent the average for a heating season.

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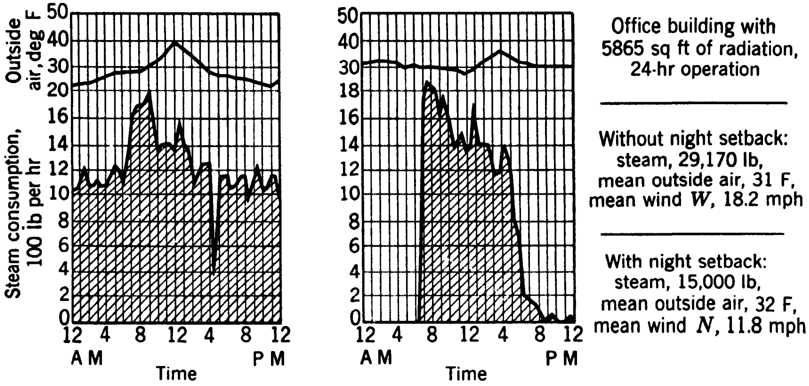
[3] "Operation of the Research Home with Reduced Room Temperature at Night," by A. P. Kratz, W. S. Harris, and M. K. Fahnestock, *ASHVE, Trans.* Vol. 49, 1943.

Important savings in either the fuel requirements of isolated plants or in the amount of purchased steam may be obtained by night setback operation, the extent of which is dependent upon the minimum practical night temperature and the hours of occupancy. The restoration of a normal daytime temperature required a peak load of 1850 lb of steam per hr, following night shut off, as indicated by the right-hand steam-consumption curve of Fig. 375, while with continuous steam consumption the morning peak load was 1780 lb per hr as indicated by the left-hand curve. These items are an indication of plant-capacity requirements and tend to bear out the fact that small isolated plants may require a considerable time of operation with heavy overloads and possibly reduced efficiency of operation to restore normal conditions. However, the situation is generally such that all savings produced by operation with periods of reduced air temperatures are not lost.



Office building with 5865 sq ft of radiation, 24-hr operation

Without night setback: steam, 29,170 lb, mean outside air, 31 F, mean wind *W*, 18.2 mph

With night setback: steam, 15,000 lb, mean outside air, 32 F, mean wind *N*, 11.8 mph

FIG. 375. Steam consumption of a building with and without night shut-off.

TABLE 132

Areas and Circumferences of Circles

Diameter	Area	Circumference	Diameter	Area	Circumference	Diameter	Area	Circumference
⅛	0.0123	.3927	16	201.06	50.265	54	2290.2	169.646
¼	0.0491	.7854	½	213.82	51.836	55	2375.8	172.788
⅜	0.1104	1.1781	17	226.98	53.407	56	2463.0	175.929
½	0.1963	1.5708	½	240.52	54.978	57	2551.7	179.071
⅝	0.3067	1.9635	18	254.46	56.549	58	2642.0	182.212
¾	0.4417	2.3562	½	268.80	58.119	59	2733.9	135.354
⅞	0.6013	2.7489	19	283.52	59.690	60	2827.4	188.496
1	0.7854	3.1416	½	298.64	61.261	61	2922.4	191.637
⅛	0.9940	3.5343	20	314.16	62.832	62	3019.0	194.779
¼	1.227	3.9270	½	330.06	64.403	63	3117.2	197.920
⅜	1.484	4.3197	21	346.36	65.973	64	3216.9	201.062
½	1.767	4.7124	½	363.05	67.544	65	3318.3	204.204
⅝	2.073	5.1051	22	380.13	69.115	66	3421.2	207.345
¾	2.405	5.4978	½	397.60	70.686	67	3525.6	210.487
⅞	2.761	5.8905	23	415.47	72.257	68	3631.6	213.628
2	3.141	6.2832	½	433.73	73.827	69	3739.2	216.770
¼	3.976	7.0686	24	452.39	75.398	70	3848.4	219.911
½	4.908	7.8540	½	471.43	76.969	71	3959.2	223.053
¾	5.939	8.6394	25	490.87	78.540	72	4071.5	226.195
3	7.068	9.4248	26	530.93	81.681	73	4185.3	229.336
¼	8.295	10.210	27	572.55	84.823	74	4300.8	232.478
½	9.621	10.996	28	615.75	87.965	75	4417.8	235.619
¾	11.044	11.781	29	660.52	91.106	76	4536.4	238.761
4	12.566	12.566	30	706.86	94.248	77	4656.0	241.903
½	15.904	14.137	31	754.76	97.389	78	4778.3	245.044
5	19.635	15.708	32	804.24	100.531	79	4901.6	248.186
½	23.758	17.279	33	855.30	103.673	80	5026.5	251.327
6	28.274	18.850	34	907.92	106.814	81	5153.0	254.469
½	33.183	20.420	35	962.11	109.956	82	5281.0	257.611
7	38.484	21.991	36	1017.8	113.097	83	5410.6	260.752
½	44.178	23.562	37	1075.2	116.239	84	5541.7	263.894
8	50.265	25.133	38	1134.1	119.381	85	5674.5	267.035
½	56.745	26.704	39	1194.5	122.522	86	5808.8	270.177
9	63.617	28.274	40	1256.6	125.664	87	5944.6	273.319
½	70.882	29.845	41	1320.2	128.805	88	6082.1	276.460
10	78.54	31.416	42	1385.4	131.947	89	6221.1	279.602
½	86.59	32.987	43	1452.2	135.088	90	6361.7	282.743
11	95.03	34.558	44	1520.5	138.230	91	6503.8	285.885
½	103.86	36.128	45	1590.4	141.372	92	6647.6	289.027
12	113.09	37.699	46	1661.9	144.513	93	6792.9	292.168
½	122.71	39.270	47	1734.9	147.655	94	6939.7	295.310
13	132.73	40.841	48	1809.5	150.796	95	7088.2	298.451
½	143.13	42.412	49	1885.7	153.938	96	7238.2	301.593
14	153.93	43.982	50	1963.5	157.080	97	7389.8	304.734
½	165.13	45.553	51	2042.8	160.221	98	7542.9	307.876
15	176.71	47.124	52	2123.7	163.363	99	7697.7	311.018
½	188.69	48.695	53	2206.1	166.504	100	7854.0	314.160

TABLE 133

SYMBOLS FOR DRAWINGS—DUCTWORK, DAMPERS, GRILLES, AND CONNECTIONS

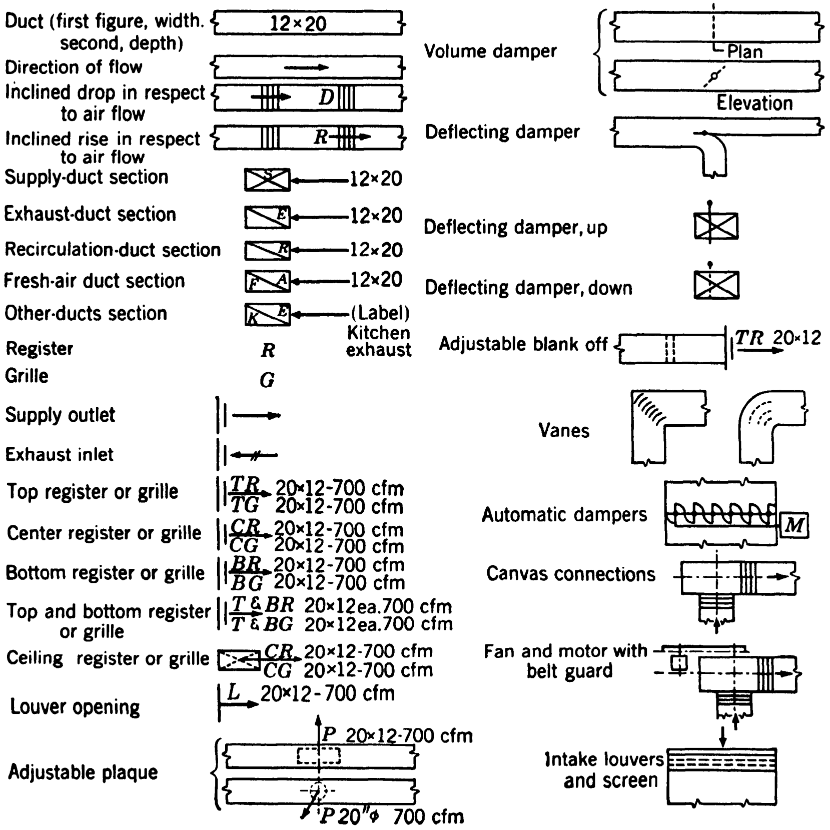


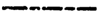
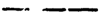
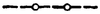
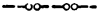
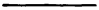
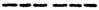
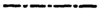
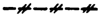
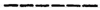
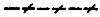
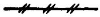
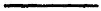
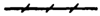
TABLE 134

Symbols for Drawings—Piping, Traps, Valves, and Refrigeration

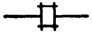
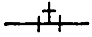
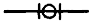
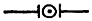
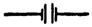
Air-Conditioning and Refrigeration Piping

Brine return	— — *BR* — —
Brine supply	——*B*——
Circulating chilled or hot-water flow	——*CH*——
Circulating chilled or hot-water return	——*CHR*——
Condenser water flow	——*C*——
Condenser water return	— — *CR* — —
Drain	——*D*——
Humidification line	—··*H*··—
Refrigerant, discharge	——*RD*——
Refrigerant, suction	— — *RS* · —

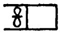
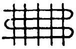
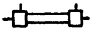
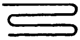
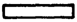
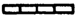
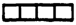
Heating Piping

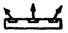
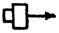
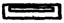
Air-relief line	
Boiler blow-off	
Compressed air	——*A*——
Condensate or vacuum-pump discharge	
Feedwater pump discharge	
Fuel-oil flow	——*FOF*——
Fuel-oil return	— — *FOR* — —
Fuel-oil tank vent	——*FOV*——
Hot-water heating supply	
Hot-water return	
Make-up water	
Return, high-pressure	
Low-pressure	
Medium-pressure	
Steam, high-pressure	
Low-pressure	
Medium-pressure	

Fittings

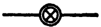
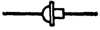
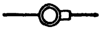
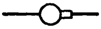
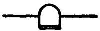
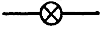
Elbow	
Elbow, looking down	
Elbow, looking up	
Flanges	
Tee	
Tee, looking down	
Tee, looking up	
Screwed union	

Heat-Transfer Apparatus

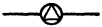
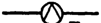
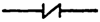
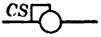
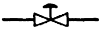
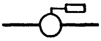
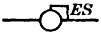
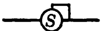
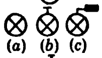
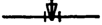
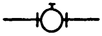
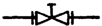
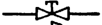
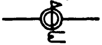
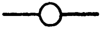
Condenser, evaporative	
Cooling unit, immersion	
Forced convection	
Finned-type natural convection	
Cooling tower	
Heat interchanger	
Pipe coil	
Radiator, plan	
Wall radiator, plan	
Wall radiator, on ceiling, plan	

Unit-heater (centrifugal fan) plan	
Unit-heater (propeller) plan	
Unit ventilator plan	

Traps

Blast, thermostatic	
Boiler return	
Float	
Float and thermostatic	
Scale	
Thermostatic	

Valves

Air eliminator	
Air line	
Angle	
Stem perpendicular	
Check	
Compressor suction pressure limiting, Throttling-type (Compressor side)	
Diaphragm	
Evaporator, pressure-regulating, thermostatic, throttling-type	
Evaporator, pressure-regulating, throttling-type (evaporator side)	
Evaporating, pressure-regulating, snap action	
Expansion, automatic (*a*) hand (*b*) thermostatic (*c*)	
Gate	
Globe	
Hand shut-off, flanged	
Lock and shield	
Reducing pressure	
Solenoid	

Miscellaneous Details

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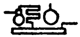
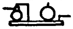
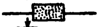
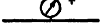
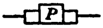
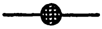
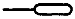
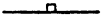
Condensing unit, air cooled	
Water cooled	
Dryer	
Float, high side	
low side	
Gage	
Switch, pressure with high-pressure cut-out	
Strainer	
Thermal bulb	
Thermometer	
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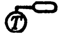
Thermostat, with remote bulb	

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